Quenching steels with gas jet arrays

PAUL F STRATTON
ANDREW P RICHARDSON
BOC
Rother Valley Way, Holbrook, Sheffield
UNITED KINGDOM
Paul.stratton@boc.com http://www.catweb.boc.com

Abstract: - Single components heated in vacuum or conventional atmosphere furnaces can be quenching by gas jets at higher quenching rates than those possible with conventional multi-component gas quenching. Such treatments may be seen as meeting the need for a clean, non-toxic quenching medium that leaves no residues to be removed after processing. The processing of single components allows the operator complete control of the quenching intensity both locally and generally. Moreover the quenching rate may be changed during the quenching cycle.

In order to achieve the best system performance it is necessary to optimise the performance of the jet array. Two-dimensional CFD was used to model and optimise the distance from jet to workpiece, the distance between jets and the gas velocity, and the mean heat transfer coefficient was calculated. A test rig to quench a gear blank was constructed using the optimised array. This rig was then modelled in three dimensions and the results compared to those generated experimentally. Some modifications to the model were necessary to align it with the experimental results.

Keywords: - Quenching, Nitrogen, Jet arrays, Modelling, Validation, Steel

1 Introduction

Gas quenching in vacuum furnaces has been used for several years, usually under nitrogen, argon or helium at pressures up to 60 bar, and its characteristics for bulk quenching of components are well known [1]. It has recently been suggested that gas quenching could be applied to single components or small groups that were heated in either vacuum or conventional atmosphere furnaces [2]. To eliminate the need to cool the furnace structure, these techniques often require the transfer of the component to be quenched to a specially designed cold chamber [3].

Gas jet quenching applied to a single component heat treated in either vacuum or conventional atmosphere furnaces can achieve higher quenching rates than would be possible with conventional multi-component gas quenching [4]. Such treatments may be seen as meeting the need for a clean, non-toxic quenching medium that leaves no residues to be removed after processing. The processing of single components using a jet array allows the operator almost complete control of the quenching intensity both locally and generally. Moreover the quenching rate can be changed during the quenching cycle. It may therefore be possible to marquench one area of a component and fast oil quench another in a single operation.

If the benefits of gas quenching are to expand beyond the niche process areas where its high uniformity and high repeatability justify a premium price, then a process that uses the minimum quantity of the lowest possible cost gas is required. It is an unfortunate fact that the high conductivity gases such as helium and hydrogen are expensive. The low cost alternative, nitrogen, has poor thermal performance. However, it has been shown that the delivery pressure available from liquid nitrogen storage systems is able to generate the jet velocities required to produce at least oil-like quenching characteristics [4].

2 Optimising the jet array

In order to meet the criteria for uniform quenching of a single component, the quenchant has to cool the surface of the component uniformly. For a gas to achieve this aim, it must have the same speed and direction at all points although in practice it has to enter through discrete nozzles. Modelling shows that, under most conditions, the cooling is at a maximum directly in line with the nozzle and falls away to the mid-point between the nozzles, as shown in Fig. 1.

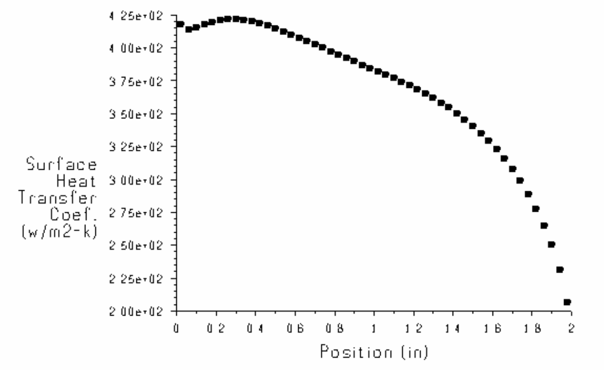

Fig. 1. The surface heat transfer coefficient for v=100 m/s, a= 50.8 mm (2 inches) and b=88.9 mm (3.5 inches).

The quench process was modelled using the Fluent v5.0 computational fluid dynamics software package. For this initial screening only two dimensions were modelled to speed up the process. The model consisted of an array of gas nozzles 12.7 mm (0.5 inches) in diameter perpendicular to the hot surface. The distance between the nozzles and the surface (a) and the distance between nozzles themselves (b) was varied for a range of imposed gas velocities (v) at the exit from the nozzle. A typical velocity profile derived from the model is shown in Fig. 2. A closer view of the velocities at the surface (Fig. 3) shows that the flow over the major part of the surface is far from the optimal perpendicular and is in fact parallel to it, reducing the maximum heat extraction rate.

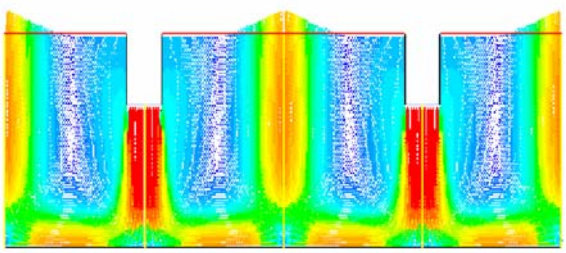

Fig. 2. A typical velocity profile for v=100 m/s, a= 50.8 mm (2 inches) and b=88.9 mm (3.5 inches).

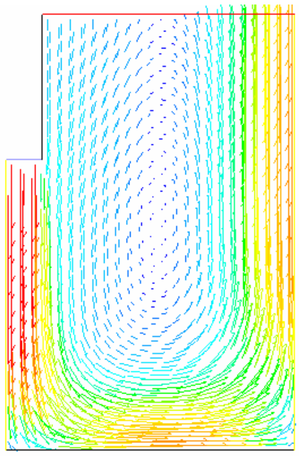

Fig. 3. A close up view of the velocity profile for v=100 m/s, a= 50.8 mm (2 inches) and b=88.9 mm (3.5 inches).

The heat transfer coefficient for the hot surface was calculated as a function of the distance from the centre line of the nozzle. The surface heat transfer profile for each set of conditions was integrated to give the average heat transfer coefficient. These values were plotted as a function of distance between nozzles and the surface (a) in Fig. 4.

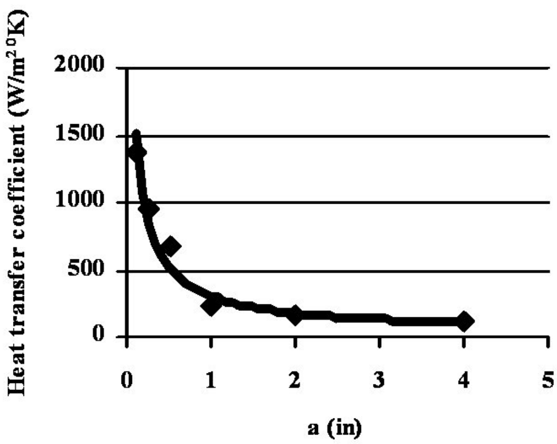

Fig. 4. The variation of mean surface heat transfer coefficient with the distance between the surface and the nozzles (a) for v=100 m/s and b=12.7 mm (0.5 inches)

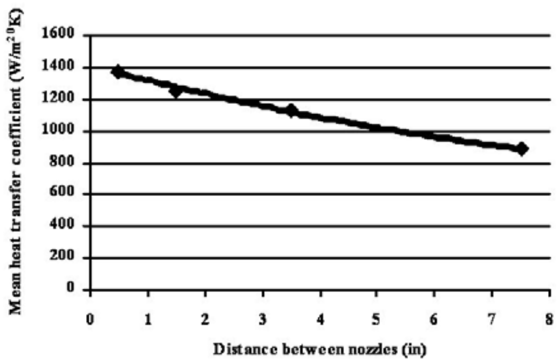

Fig. 5. The variation of mean surface heat transfer coefficient with the distance between the nozzles (b) for v=100 m/s and a=3.2 mm (0.125 inches)

To keep down costs it is obviously necessary to minimise gas flow. As the gas flow for a given nozzle is fixed by the cooling rate required, the only variable available is the distance between nozzles. Somewhat surprisingly, the distance between the nozzles has little effect on the heat transfer coefficient as can be seen from Fig. 5. This effect is due to the area of high turbulence created at the edge of the nozzle at high gas velocities. This effect can clearly be seen in the velocity vector diagram (Fig. 6).

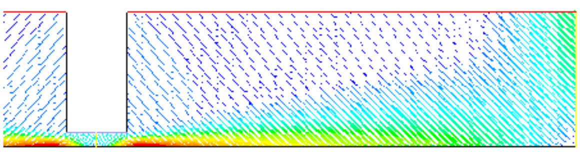

Fig. 6(a). The velocity vector diagram for a nozzle to surface distance of 3.2 mm (0.125 inches), distance between nozzles of 190.5 mm (7.5 inches) and a gas velocity of 100 m/s.

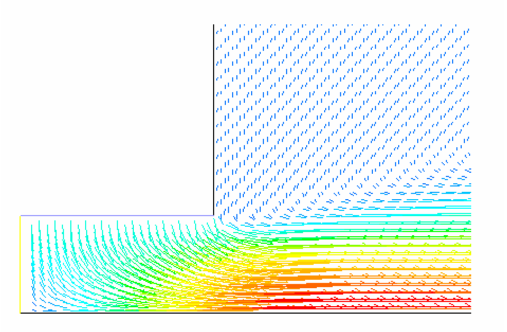

Fig. 6(b). A detail close to the nozzle edge from the velocity vector diagram for a nozzle to surface distance of 3.2 mm (0.125 inches), distance between nozzles of 190.5 mm (7.5 inches) and a gas velocity of 100 m/s.

The heat transfer coefficient is also relatively insensitive to scaling factors; i.e. if all the size factors are reduced by a factor of four, which is likely to include the maximum practical range of jet sizes, there is only a 30% increase in the heat transfer coefficient [5].

This lack of sensitivity to the size of the nozzles and the distance between them simplifies the design of quenching enclosures, especially for complex shapes. However the close approach to the surface required by the technique does result in the need for careful consideration of the nozzle sites. As a result of the high gas pressures, it should be possible to eliminate the need to support the product during quenching. The effect of the product's weight will be small compared with the applied force of the gas and the product would float within the nozzle field. If the ratio of the diameter of the nozzle and the distance between the nozzle and the surface is chosen as four (the point at which the area for gas escape equals the area of the nozzle), then the system is self-compensating. This is because any reduction in distance will increase the pressure at the nozzle for a given flow and increase the separation again. Small inconsistencies would be introduced into the flow field in a practical device and would lead to oscillation or rotation of the component, which would produce more even quenching. The high velocities used will probably lead to high noise levels in the vicinity of the quench, but this effect could be minimised by sound insulation around the cold wall quenching chamber.

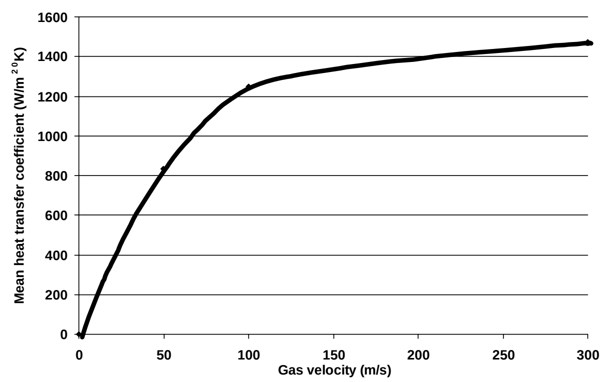

Fig. 7. The effect of velocity on mean heat transfer coefficient for a=3.2 mm (0.125 inches) and b=38 mm (1.5 inches)

The cooling rate is almost directly proportional to the gas velocity at gas velocities below 100 m/s (Fig. 7), and the velocity is related to the supply pressure. It is obviously simple, therefore, to control the cooling rate. Although very high velocities towards sonic will result in higher cooling rates, the rate of increase becomes non-linear, so their use is likely to be restricted to applications where the highest possible cooling rates are required.

3 Validating the model

In order to validate the model it was necessary to construct a test rig. When the rig had been constructed it was modelled with Fluent 6.0.12 using the same conditions that had been used in earlier work (Stratton et al, 2000), except that some of the physical characteristics of the steel were replaced with more recent data.

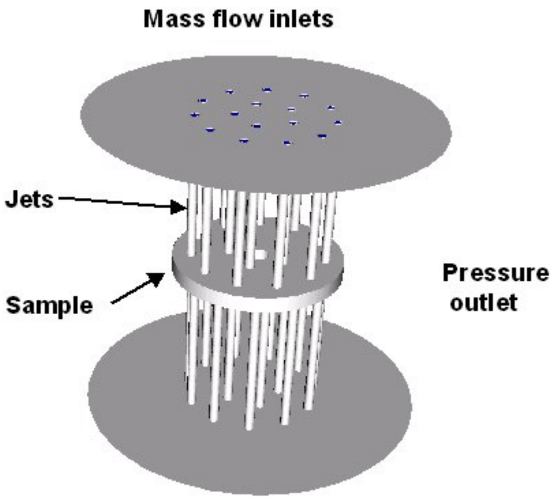

Fig. 8. The model

The model domain was set at a 200 mm radius, almost twice the radius of the specimen, and extended vertically to the gas distribution manifolds above and below the sample (200 mm total). The specimen and tube bank arrays to form the jets were thus centrally located within this cylindrical domain (Fig. 8).

The model was meshed in two parts. The internals of the gas feed tubes and specimen itself were meshed using a regular hexahedral scheme (50,000 cells), while the gas space was meshed using a pyramidal scheme (800,000 cells). Attention was placed particularly on resolving the mesh near the tube tips and specimen surface. Because different mesh densities were needed in the specimen and in the gas flow space, non-conformal interfaces were set up on the specimen boundaries. This allowed higher quality meshes with fewer cells to be generated for each region.

Initially the standard Fluent segregated solver was used together with first order discretisation for momentum, energy and turbulence parameters; nitrogen was simulated as an incompressible gas. The k-e turbulence model was also used with standard wall functions. An initial steady solution was achieved for a cold flow and then for a fixed hot sample temperature. During this period the mesh at the tips of the nozzles and on the surface of the sample were repeatedly adapted to ensure that the boundary layer assumptions of the model were within the valid range (Y^+ in the range 30-60).

The solver was then switched to the unsteady mode (1st order implicit) with a time step of 1 second. Although the maximum number of iterations per second was set to 100 to ensure a good degree of convergence at each time step, the model would typically converge with considerably fewer iterations, especially towards the end of the simulation. Convergence criteria were set for normalized unscaled residuals of 10^{-3} for continuity, velocity and turbulence and 10^{-6} for energy and radiation, with model mass and energy imbalances monitored periodically across boundaries. Calculations performed on a dual 1.7 GHz P4 processor Dell Precision workstation with 1 GB RAM took approximately 90 seconds per iteration. Because of time constraints and the close correlation with experimental results this work did not examine at the effects of various modeling options, e.g. higher order discretisation schemes, turbulence models and compressibility.

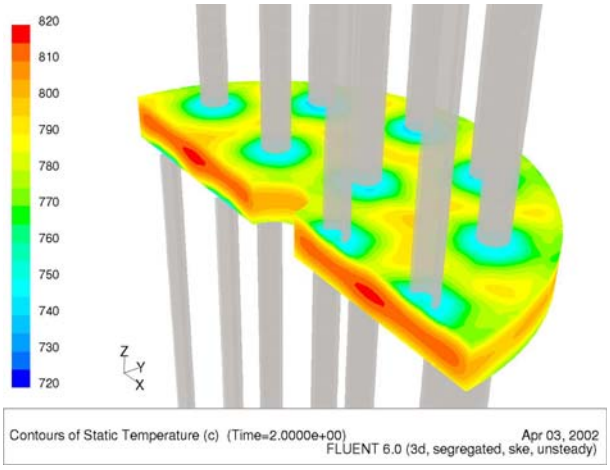

Fig. 9. The temperature distribution after 2 seconds

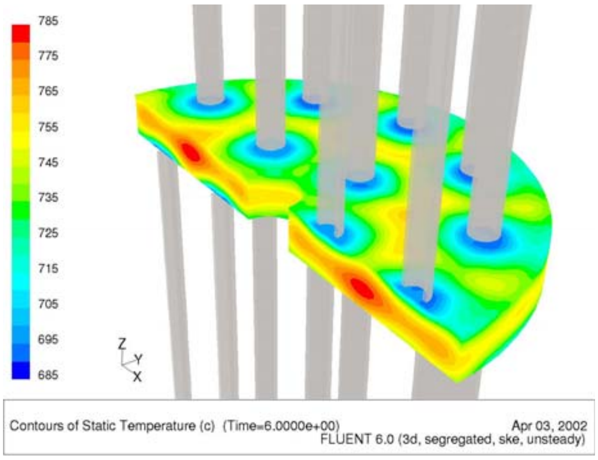

Fig. 10. The temperature distribution after 6 seconds

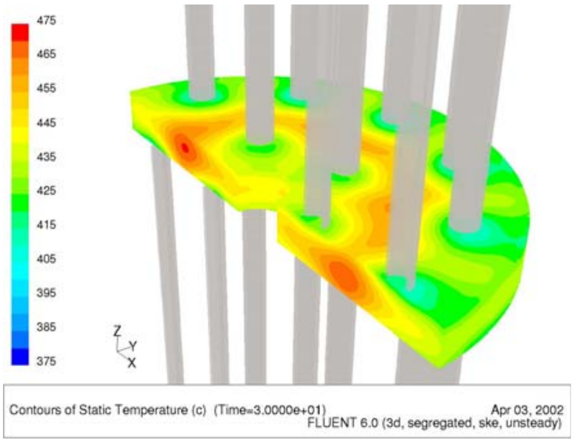

Fig. 11. The temperature distribution after 30 seconds

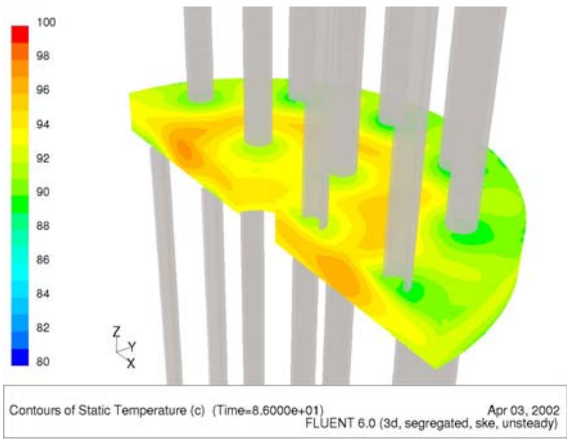

Fig. 12. The temperature distribution after 86 seconds

Typical model outputs for 2, 6, 30 and 86 seconds quenching are shown in Figs. 9 to 12 respectively. The model was also used to predict the cooling curve at the position of the thermocouple.

Several experimental runs were carried out. The results (Fig. 13) show clearly that the model predicted a much lower initial cooling rate than occurred in practice. Fig. 14 compares the actual cooling rate with that predicted by the model. The large difference in initial cooling rate was probably caused by radiative heat losses that were not taken into consideration in the model.

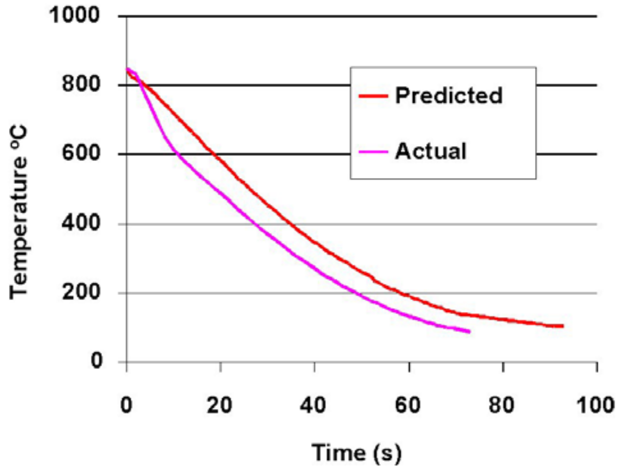

Fig. 13. Actual and predicted cooling curves

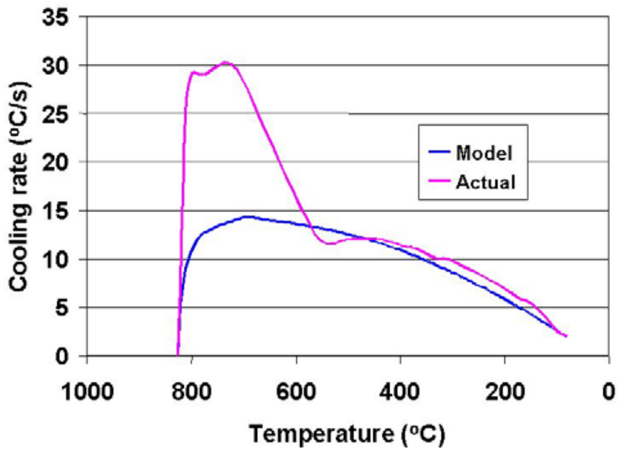

Fig. 14. Comparison of the actual cooling rate with that predicted by the model

The modelling was therefore repeated taking radiative heat losses into account. The effect of this modification is shown in Fig. 15. The discrete-ordinates model was chosen to simulate the radiative losses with a sample surface emissivity of 0.8. The simulation was started from the actual pre-heat furnace temperature to allow for a more representative thermal distribution at the start of the gas quench. This choice of assumed emissivity was validated, as the modelled cooling by radiation was at a similar rate to that measured in the experiment. There is now a significant difference between the cooling rates at the surface and in the core of the sample, due to the high initial heat loss from the surface by radiation. The modelled cooling rate in the core is increased over the same sort of temperature range as the actual but not to the same magnitude.

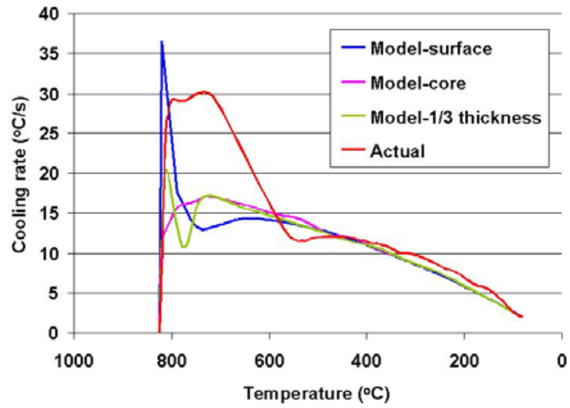

Fig. 15. Comparison between actual and modelled cooling rates when radiation is taken into account

It was suggested that the equilibrium specific heat capacity (Cp) values used for the model were inappropriate for continuous cooling conditions [6]. Cp values appropriate to the phases present at the time derived from the continuous cooling curve were substituted with the results shown in Fig. 16. This gave an almost perfect match with the experimental data.

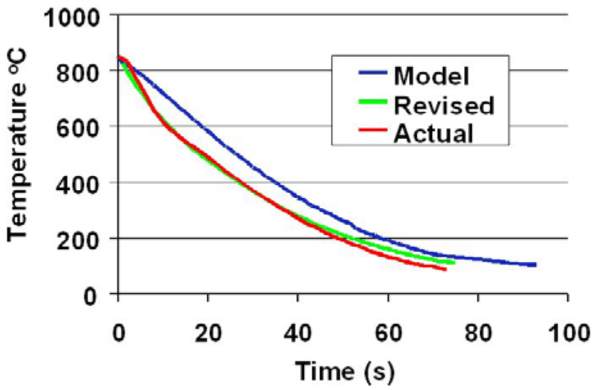

Fig. 16. Comparison of the original and revised model quenching curves with the experimental data

4 Conclusions

Jet arrays for quenching steel components can successfully be optimised by modelling them using CFD. An array with the jets about four to eight times their own diameter apart and the distance of a quarter of the diameter from the surface with a jet velocity of 100 m/s was found to be optimum. The results of experimental work suggested that radiative heat loses must be included in the model. It also showed the importance of using dynamic rather than equilibrium data for modelling steels under a rapidly changing temperature regime.

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