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An experimental investigation of the dynamic behaviour of a shell-and-tube condenser

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Abstract—Experimental data are presented on the transient behaviour of an industrial scale shell-and-tube exchanger condensing a mixture of steam and air under reduced pressure. The system was subjected to step changes in the five key loads which determine the condenser behaviour. These are the inlet pressure, steam flowrate, air flowrate, coolant flowrate and coolant inlet temperature. The experimental equipment and its associated instrumentation are described. The experiments performed are detailed and the results from two representative runs are presented and discussed. An understanding of the process is a necessary pre-cursor to modelling and simulation. © 1997 Elsevier Science Ltd.

INTRODUCTION

Condensation is an important unit operation in the chemical process industries. It is closely associated with distillation, evaporation and crystallisation processes. It would appear that the transient behaviour of industrial condensers has not previously been examined in detail. The behaviour under such conditions has safety, control, economic and environmental implications. Questions such as how the system will respond to a change in flowrate, temperature, pressure or composition, need to be addressed. The system's response to, for example a failure in the coolant supply, a sudden surge of vapour feed or loss of vacuum, is of great interest. There are also normal operating conditions where the dynamic behaviour is important: in process control, plant start-up and shutdown for example. The inability of a condenser to meet the required duty under any one of these circumstances leads to incomplete condensation. Economically and perhaps environmentally damaging losses may result in the case of a condenser vented to atmosphere.

In this work an industrial-scale condenser, installed in the Pilot Plant of the Department of Chemical Engineering at UMIST, was instrumented such that the dynamic response of the system could be seen. The five key process loads that determine the condenser behaviour are pressure, steam flowrate, air flowrate, cooling water flowrate and cooling water temperature. To each of these a step change in value was imposed and the ensuing approach to a new steady state observed. From the results it is possible to distinguish the physical phenomena that accompany these changes. The system studied was the condensation of steam from air under reduced pressure.

This study into the transient behaviour of condensers is part of a joint program of work undertaken by the departments of Chemical Engineering at Stuttgart and UMIST. On the basis of the experimental work a model of a shell-and-tube condenser has been developed within the dynamic process simulation environment, DIVA [1], and this has been used to simulate the test condenser. The development of a successful condenser model requires detailed and reliable experimental data in order to validate its behaviour and predictions.

EXPERIMENTAL SYSTEM

The industrial–scale condenser in the Chemical Engineering Department pilot plant at UMIST has been in operation since 1982. It has been used for a number of research projects including studies at both atmospheric and reduced pressure with steam–air mixtures and mixed hydrocarbons [2–4]. These were primarily concerned with the performance of the equipment at steady-state. The system was modified to allow the unsteady performance of the condenser to be investigated [5].

The experimental system consists of the condenser

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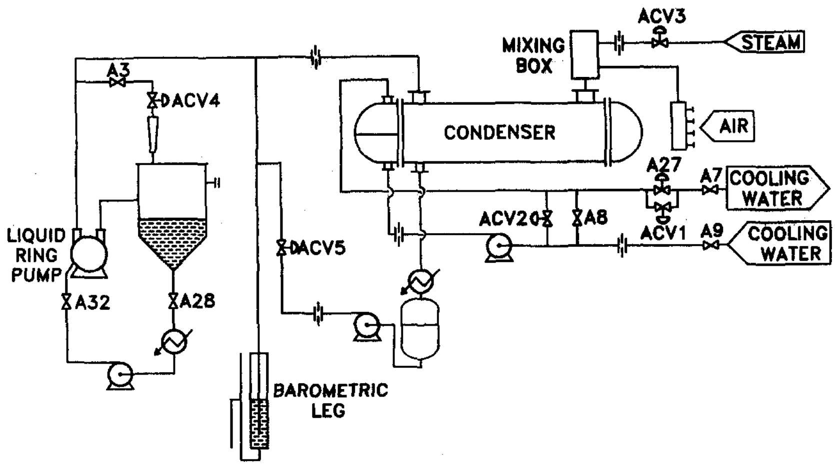

Fig. 1. Simplified flowsheet of the experimental system.

itself, a cooling water circuit, a vacuum pump and the associated plant instrumentation. A simplified flowsheet is shown in Fig. 1. The condenser is a TEMA [6] 'E' shell-and-tube heat exchanger of 0.438 m inside diameter and an effective tube length of 2.438 m. The main geometrical features of the condenser are shown in Fig. 2. Condensation takes place on the shell-side and cooling water flows through the tubes, which are of 19.05 mm diameter, in a horizontally divided two pass arrangement with 98 active tubes per pass. The tubes are on a 25.4 mm triangular pitch with a characteristic angle of 60°. The shell-side is divided into eight compartments by seven vertically cut segmental baffles with a baffle cut of 35.5%. The baffles are extended below a horizontal chord 50 mm above their lowest point and that sector is welded to the shell thus forming eight liquid tight compartments. The exchanger is mounted at an angle of 2.4° to the horizontal to facilitate condensate drainage. The cooling water turnaround header is fitted with a horizontal baffle to produce a narrow window between the baffle and the end wall. The high velocities induced in the flowing liquid ensure efficient turbulent mixing. The vapour feed is a mixture of steam and air. This enters horizontally through a 0.203 m diameter nozzle with no impingement plate fitted.

Steam at 5 barg is provided from the plant main and loads of up to 1 MW are possible. The steam flowrate is controlled by pressure upstream of a nozzle in critical flow by automatic valve ACV3 in Fig. 1. Air enters via one of a set of calibrated nozzles in critical flow, giving mass flowrates from 5 to 175 kg h^{-1}, corresponding to air–water mole ratios up to 0.1. These are mixed in an expansion box at the inlet.

A liquid ring pump with a nominal capacity of 0.1 $m^3\ s^{-1}$ is used to evacuate the condenser and associated equipment to a pressure substantially below atmospheric and maintain the vacuum level despite air leakage. Condenser pressure is controlled by a recycle flow from pump discharge to suction by automatic valve ACV4.

The condensate drains to the baffle space at the condenser outlet. It then passes via a cooler to a buffer tank. This acts as a supply reservoir for the condensate pump and provides a means of regulating the condensate flowrate, where automatic valve ACV5 acts

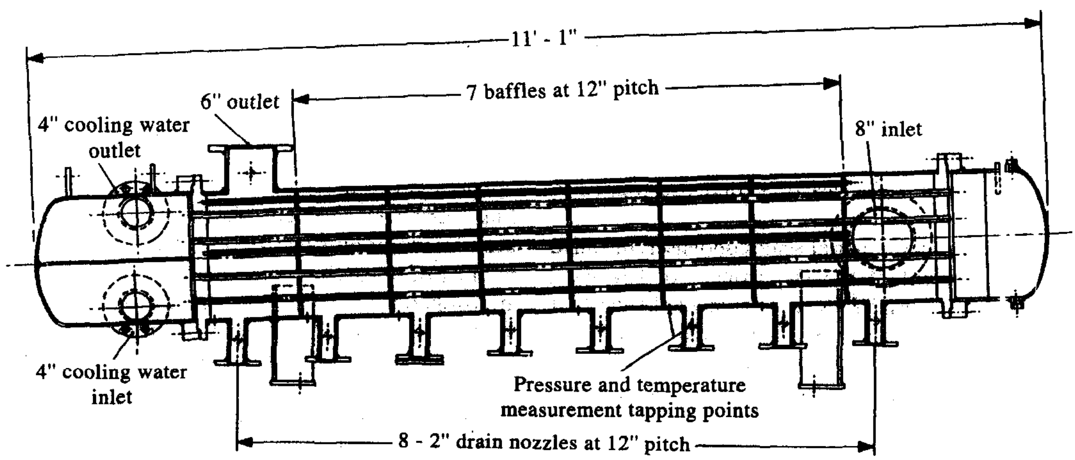

Fig. 2. Test condenser.

to control the tank level. The condensate then flows to a barometric leg and from there is discharged to drain.

Cooling water is supplied from the site services. The water is circulated around a closed loop at a rate controlled by valve ACV2 with the provision for fresh coolant to be admitted at a rate controlled by valve ACV1 in order to achieve the desired cooling water inlet temperature. Flowrates of up to 28 kg s^{-1} can be achieved resulting in average tube velocities of up to 1.8 m s^{-1}. It is unfortunate that the control of cooling water flowrate and inlet temperature interact with one another and a careful strategy is needed to produce good step changes.

The apparatus is fully instrumented so that vapour, coolant and condensate flowrates and temperatures and system pressures can be measured continuously. The temperature profile is measured by nine thermocouples located at the inlet, opposite each baffle and at the outlet. The shell-side pressure profile is measured at the inlet, opposite the first, second, third and fifth baffles and at the outlet. There are more pressure gauges close to the inlet as the vapour flowrate and hence pressure drop is greatest in this region. Data logging is accomplished using a microcomputer controlled interface. The software used enables sampling at frequencies of up to 25 kHz. The data produced are converted to a format which may be read into commercial spreadsheet packages.

EXPERIMENTAL WORK

The behaviour of the condenser is determined by the values of five key loads; inlet pressure, steam flowrate, air flowrate, coolant flowrate and coolant inlet temperature. To each of these quantities a step increase and a step decrease were imposed in order to obtain the unsteady response of the system. The other four conditions were kept as constant as possible, allowing the response to the changed variable alone to be observed.

The experimental system is interfaced with a process control computer which enabled the operation of a number of automatic control loops and the setting of control valves to any desired position. Step changes in all the key variables but air flowrate were achieved remotely. The air load was changed by physically removing or replacing rubber flaps which seal the nozzle box orifices. The system temperatures and pressures were recorded by the logging device at a frequency of 2 Hz. Recording commenced 2 min before the step change was imposed and was continued until the new steady state was achieved.

For each variable a step increase and a corresponding step decrease were imposed. The experiments performed were as follows:

- step increase and decrease in condenser outlet pressure between 0.137 and 0.395 bar;
- step increase and decrease in steam flowrate between 11.1 and 18.1 kg s^{-1};
- step increase and decrease in air flowrate between 29.9 and 120.6 kg h^{-1};
- step increase and decrease in cooling water flowrate between 19.4 and 23.1 kg s^{-1};
- step increase in cooling water inlet temperature between 25.1 and 29.1°C and step decrease between 32.1 and 27.1°C.

In order to confirm the reliability of the measurements the overall mass and energy balances of the system were calculated at the initial and final steady-states. For the whole series of experiments the energy balances were within $\pm 4\%$.

The results for a step increase in condenser outlet pressure and a step decrease in steam flowrate are given in full. In interpreting the results it is necessary to understand that operating condensers are underloaded. A pocket of air will accumulate near the vent. The temperature of the vapour will show a characteristic shape. At and near the inlet the condensing temperature will lie close to the saturation temperature of the vapour, and near the outlet the temperature approaches the coolant temperature. A *condensation front* lies between and is identified by a rapid change of temperature with position and often a characteristic increase in the noise of the measured temperatures and pressures. It separates the regions of almost isothermal pure condensation and a region where gas cooling becomes increasingly important.

Step increase in condenser outlet pressure

In this experiment the condenser outlet pressure was increased from 0.137 to 0.395 bar and the measured inlet and outlet pressures are shown in Fig. 3. The measured air flowrate was held constant at 28.3 kg h^{-1} as determined by critical flow through a nozzle. Measurement shows only a slight fluctuation in the inlet coolant temperature of 36°C, Fig. 4, and the constancy of the steam and coolant flowrates at 12.4 kg min^{-1} and 23.1 kg s^{-1}, respectively, Fig. 7.

An increase in operating pressure causes the saturation temperature to rise, $T_{sat} \sim 53$°C at 0.14 bar, $T_{sat} \approx 76$°C at 0.4 bar, and the molar holdup of vapour

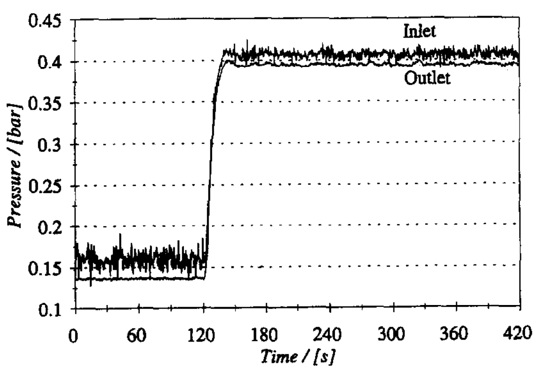

Fig. 3. Step increase in pressure.

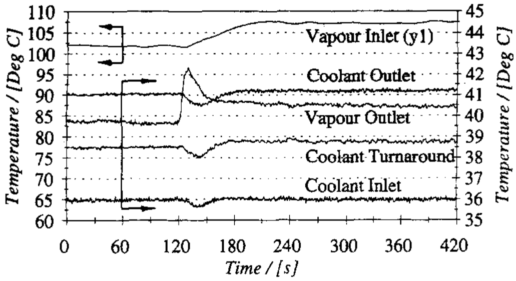

Fig. 4. Responses to an increase in pressure.

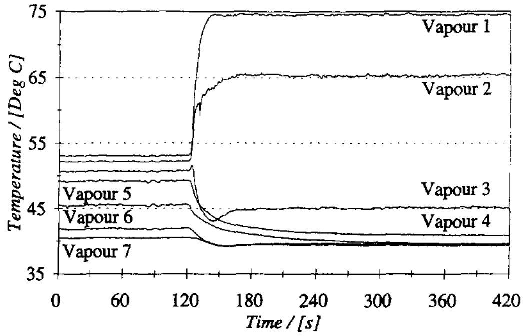

Fig. 5. Vapour responses to an increase in pressure.

in the condenser to increase. The change in saturation temperature causes an increase in the temperature driving force with a corresponding decrease in the condensation area required. More complete condensation is achieved but somewhat paradoxically the outlet temperature of the vapour is higher. Further the increase in vapour holdup leads to a temporary reduction in the condensation rate. The effects these changes have on the system can be seen in the responses of the pressures, temperatures and flowrates.

The pressure plot, Fig. 3, shows that a good step change in outlet pressure was achieved. The pressure drop over the condenser is lower after the step increase in pressure, as is clear from the reduced separation between the two curves. The pressure drop is proportional to $\int \rho u^2 \,dl$ and both the length, l (l being the downstream position of the condensation front) and the vapour velocity, u, are reduced. This is because the condensation front moves towards the inlet and at the higher pressure the velocity is smaller and the vapour condenses more quickly.

The cooling water inlet and outlet show small dips in temperature after the increase in pressure, Fig. 4. This is due to a temporary drop in condensation rate caused by the greater vapour holdup at this higher pressure. The dip in temperature at the cooling water inlet is smaller than that of the outlet. This is because the temperature is under automatic control which acts to maintain the inlet temperature at its set point and because the recycle flow which causes the variation is diluted by fresh feed. For all practical purposes the coolant inlet temperature may be considered constant. The coolant outlet temperature increases because the heat load is increased by the greater extent of condensation at the higher pressure.

With the increase in saturation temperature the vapour inlet temperature increases, Fig. 4. This change occurs quite slowly because the vapour is superheated before and after the step and only sensible heat is available to bring about the heating of surrounding metal walls, a gas film limited process. The vapour outlet temperature increases for the same reason but also shows a sharp peak just after the change. It then decreases somewhat as the effect of the decreased vapour temperature downstream of the *condensation front* reaches the outlet but remains above the outlet temperature before the step change. This behaviour is quite different from that of the vapour temperature opposite the last three baffles, Fig. 5, which all show lower values after the change. These effects are accounted for by a strong vertical temperature gradient with cold fresh coolant in the lower (first pass) and warmer coolant in the upper return pass. The initial surge is due to the compression of the saturated vapour present in the upper part of the condenser prior to the change. The final outlet temperature is higher by some 3.0°C than that opposite the last baffle because the vapour is heated as it passes through the hotter cooling water pass.

The *condensation front* which marks the limit of condensation within the condenser is characterised by a maximum slope of the temperature profile and is accompanied by increased fluctuations of the local temperatures. Both these features can be identified in Fig. 5. In this case the condensation area required decreases and the condensation front moves upstream. Before the step increase the temperature profile is steepest between baffles four and five indicating that the front is located in the fifth baffle space. The temperature readings at baffles four and five are also slightly noisier than the rest. After the step increase in pressure the temperature profile is steepest between baffles two and three signifying that the condensation front has moved upstream into the third baffle space. Again there is a little more noise accompanying these readings than those downstream.

The increased saturation temperature accompanying the increase in pressure causes the vapour temperatures upstream of the condensation front position, vapour 1 and vapour 2, to increase, Fig. 5. Downstream of the front, vapour 3 to vapour 7, the temperatures decrease and approach a constant value as condensation nears completion with the vapour temperature at baffles five, six and seven becoming the same. The small peak in vapour temperature opposite baffle three occurs as the condensation front moves to the third baffle space from the fifth baffle space. The subsequent temperature drop at this location is due to the abrupt decrease in condensation rate occurring there. The time constants of the temperature changes are related to that of the pressure change and also the thermal inertia of the system. The temperatures further

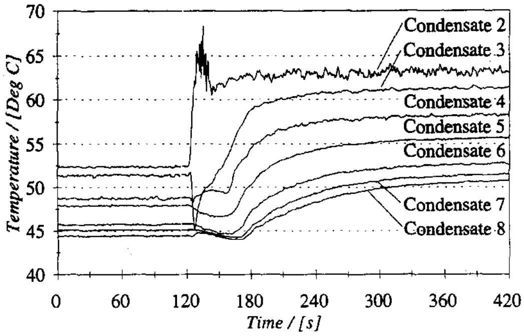

Fig. 6. Condensate responses to an increase in pressure.

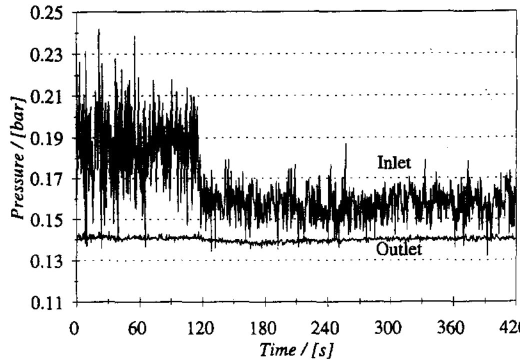

Fig. 8. Pressure responses to a decrease in steam load.

downstream decrease more slowly due to the large heat capacity of the pools of condensate and the condenser metalwork and the time it takes the condensate flow to transport this heat from the condenser.

The condensate temperatures at steady-state increase everywhere due to the increase in saturation temperature, Fig. 6. In the second baffle space the condensate temperature shows a sharp peak immediately after the change. This reflects the sudden arrival of condensate at a new higher saturation temperature and at a higher rate. The corresponding drop in condensate temperature in the third baffle space is caused by the movement upstream of the condensation front when this location experiences a sharp drop in condensation rate. For a moment only the previous, cooler condensate and a relatively small condensation load are being cooled. Shortly after the temperature increases again as the hot condensate from upstream flows in. It is clear that the responses of the condensate temperatures are progressively slower downstream.

As observed previously the flowrate plot, Fig. 7, shows that the automatic control was successful in keeping the inlet flowrates of steam and cooling water constant. However, the condensate flowrate exhibits marked oscillations. This is due to poor pressure compensation between the condenser and the condensate buffer tank which causes changes in the level difference which is used to control the rate at which the condensate is pumped from the system. Although the plot shows poor control it is linear in the flowrate out of the condenser. There is a net deficit in the condensate output over the time of the experiment. This means that the holdup of condensate within the condenser increases. This is caused by the change in location of the *condensation front*. More vapour condenses at the start of the condenser and there is an increase in the total holdup.

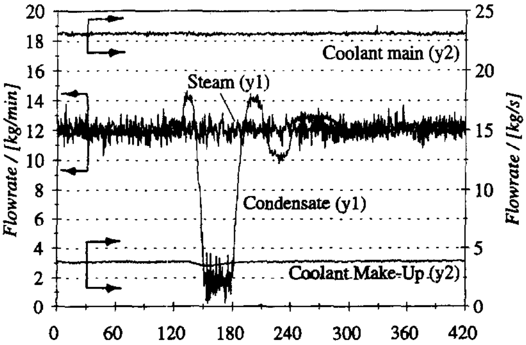

Fig. 7. Flowrate responses to an increase in pressure.

Step decrease in steam flowrate

The second experiment of which the results are shown in this paper was a step decrease in steam flowrate from 18.1 to 11.1 kg min^{-1}. The air flowrate was again held constant at 29.7 kg h^{-1} as determined by critical flow through a nozzle. Measurement shows the constancy of the condenser outlet pressure at 0.141 bar, Fig. 8, and only small fluctuations in the inlet coolant temperature from 32.1°C. Fig. 9, and the coolant flowrate from 23.2 kg s^{-1}, Fig. 12.

A decrease in steam flowrate decreases the area required for condensation and the overall pressure drop consequently decreases. With the outlet pressure held constant this leads to a lowering of the inlet pressure, Fig. 8. The decrease in pressure causes a decrease in the saturation temperature.

The cooling water inlet temperature drops immediately after the step due to the lower temperature of the recycle stream following the sharp decrease in heat load, Fig. 9. It then rises again as the control action compensates for the decrease. The cooling water outlet temperature decreases as the heat load is reduced. There is a minimum of temperature immediately after the step change caused by the corresponding dip in the cooling water inlet temperature.

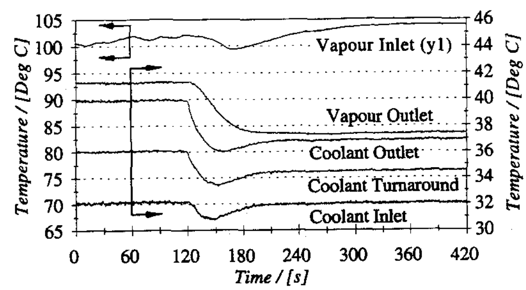

Fig. 9. Vapour and coolant responses to a decrease in steam load.

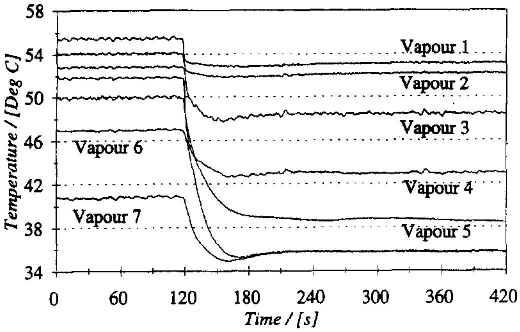

Fig. 10. Vapour responses to a decrease in steam load.

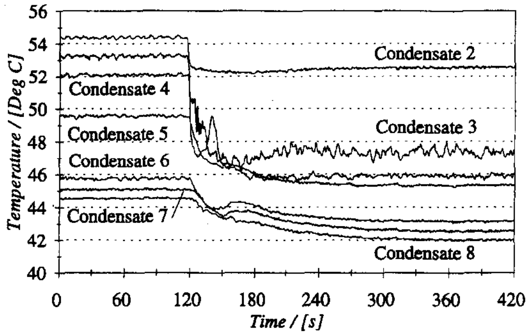

Fig. 11. Condensate responses to a decrease in steam load.

The inlet vapour temperature before the step is erratic, reflecting the fact that the steam is superheated, Fig. 9. A likely explanation is that there is some water in the bottom of the mixing box which is evaporated at an irregular rate by the superheated steam. Immediately after the step the vapour inlet temperature decreases due to the decrease in pressure and saturation temperature. It then rises again to a steady state value determined by an energy balance across the nozzle.

The vapour outlet temperature drops as a consequence of the greater degree of condensation effected and the lower temperature of the second coolant pass arising from the decreased heat load, Fig. 9. Both before and after the step change, the outlet temperature is slightly higher than the temperature opposite the last baffle, Fig. 10. As in the previous case this is because the vapour passes through the hotter coolant return pass before exiting the condenser.

As the area required for condensation decreases, the condensation front moves upstream from the sixth baffle space to the third baffle space. This is indicated by the regions of the vapour temperature profile that are steepest before and after the step, Fig. 10. The vapour temperatures decrease after the step due both to the decrease in the saturation temperature near the inlet and the overall increase in gas cooling and degree of condensation achieved. The vapour temperatures at positions that, after the step, are downstream of the condensation front, vapours 3–7, decrease the most as these positions experience the biggest change in condensation rate and vapour composition. The vapour temperatures opposite baffles six and seven recover slightly after the large initial decrease reflecting the response of the cooling water inlet temperature. The effects of the coolant temperature are more pronounced near the outlet as the condensation rate here is quite low and gas cooling predominates. The change in vapour temperatures has a longer time constant downstream because of the heat capacity of the condensate pools and of the condenser metalwork.

The condensate temperatures follow the vapour temperatures and decrease in a similar manner, Fig. 11. The greatest drop in temperature is seen for the condensate in the third, fourth and fifth baffle spaces as these experience the greatest change in condensation rate due to the relocation of the condensation front. Before the step change these positions are upstream of the condensation front, and after the decrease in steam load they are downstream of it. The small peak in condensate temperature in the fourth baffle space probably occurs as the condensate front moves through this baffle space. The time constants for the temperature change again increase downstream due to the time it takes for the warmer condensate formed before the step to run through the condenser.

The flowrate plot shows that a sharp step decrease in steam flowrate was achieved, Fig. 12. The main cooling water flowrate was kept reasonably steady with only a shallow trough after the step change. The condensate flowrate is very erratic after the step, again due to the buffer tank level control. The condensate flowrate follows that of the steam flowrate with a time lag equivalent to the residence time of the condensate in the condenser. At steady states before and after the change there is very little difference between the condensate flow out of the system and the steam flow in. The overall holdup of condensate in the condenser falls somewhat because of the decreased condensation rate and lower holdup in all the pools of condensate.

CONCLUSIONS

The results show the successful experimental measurement of unsteady condenser behaviour after

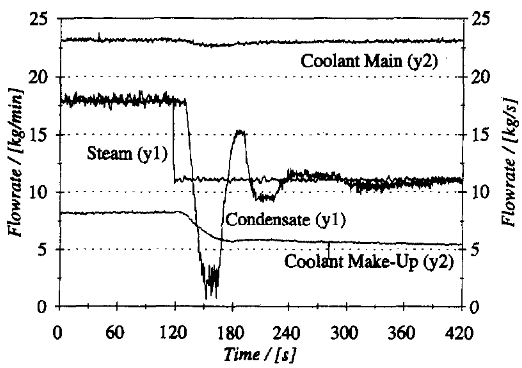

Fig. 12. Flowrate responses to a decrease in steam load.

step changes in condenser outlet pressure and steam load. These two experiments are representative of the whole set of unsteady experiments all of which yielded similar results.

The time taken for the imposed changes to be achieved are constrained by the time constants of the control equipment and the system as a whole. The pressure change has a time constant of ~ 10 s which reflects the vapour flowrates which may be limited by the capacity of the vacuum pump or by flow in the connecting pipework. The time taken for the steam step to be achieved is much faster at just over a second. The steam control valve operates very quickly and the measurement device is located immediately downstream, hence an almost instantaneous change in the steam input is observed.

Condensation is an extremely fast process with the local transfer from the bulk vapour to the interface having a time constant of the order of 0.1 s, as determined by the mass flux. Thus no effect of the fundamental process is seen in the slow system responses. The time constants of the vapour temperature changes are related to those of any pressure changes and to the heat capacity of the system. The time constants of the vapour temperature changes increase downstream due to the thermal inertia of the system and the time it takes to dissipate or absorb this heat. The response times of the condensate temperatures are slow and also increase towards the condenser outlet. The condensate flows through a series of pools each with a time constant of between 10 and 20 s. Their significant heat capacity accounts for the time it takes to reach a new steady state. The coolant residence time accounts for the long time constants of the coolant temperature responses.

The measurements taken show how the movement of the condensation front through the condenser affects the upstream and downstream vapour temperatures. The effects of a change in vapour holdup within the condenser are also observed. These bring about a transient surge or decline in condensation rate opposed to the long term consequence of the change in operating conditions.

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