TRANSIENT THERMAL STRESSES IN EMERGENCY COOLING SYSTEMS OF A REACTOR AND THE APPLICATION TO CIRCUIT INTEGRITY

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SUMMARY

In an emergency situation, cold water at 70°F is pumped via the emergency cooling system into the normal working parts of the reactor. The pipework, initially at 540°F, consequently suffers a series of thermal shocks over its lifetime in addition to the normal pressure loadings. It is necessary therefore to assess the integrity of simple parts of the system, such as plain pipes and standard Ts, under these conditions over their lifetime, before setting up a research program to investigate the more complicated parts.

The transient temperature and subsequent stress distributions were calculated by means of the finite element method, the temperatures on a triangular mesh, the nodes of which formed the isoparametric element mesh used in the stress calculations. The boundary conditions were assumed constant and a range of heat transfer coefficients taken.

The critical length of a through axial crack in a plain pipe was calculated under the pressure loadings (normal operating, maximum fault and overpressure test conditions) as it is necessary to know these before assessing the effect of the thermal shock. These revealed that the plain pipes were in a state of leak before break under pressure loading only. The overpressure test crack also did not grow to the maximum fault size over its lifetime, assuming a pessimistic value of the heat transfer coefficient.

Partial semi-elliptical axial flaws on the inside surface of the pipe, where the high tensile thermal stresses occur, were also investigated. The length of the crack was fixed at the length defined in the pressure calculations and the depth varied. Pressure and thermal loadings were considered separately as it is assumed that the pressure has dropped as a result of the accident before the thermal effects build up. The stress intensity factors, K, including edge effects, were calculated for the different geometries and compared with the reference K in ASME III. The thermal K values were calculated by the method of superposition using the finite element thermal stress distributions and the known solution for a crack in an infinite plate load at its surfaces by a pair of wedge forces.

The standard *Ts* were considered as plain pipes of thickness equal to the maximum thickness of the *T*. For the pressure stresses, a stress concentration factor of three was assumed and the subsequent K values for partial semi-elliptical defects of different depth/length ratios calculated. The thermal calculations used the superposition method and the finite element thermal stress distributions as before.

A critical size of defect was therefore established at a certain fracture toughness. The fatigue growth equation was then used to establish the minimum acceptable defect size that would grow to the critical value over its lifetime.

It was therefore established that the simple parts of the emergency cooling system were safe, under pessimistic assumptions, provided that the minimum acceptable defect size was detectable.

1. Introduction

In an emergency situation, water is pumped via an emergency cooling system into the normal working parts of the reactor. The pipe work, initially at a high temperature, consequently suffers a series of thermal shocks during its lifetime. In addition to this thermal loading, the pipework experiences the same pressure loads as the main circuit. It is therefore necessary to investigate and assess the integrity of the emergency system under these loading conditions over its lifetime. It is particular important to establish whether any water temperatures in the range of design interest give rise to an unacceptable condition.

Before setting up a work programme to investigate the more complicated parts of the emergency system, it is necessary to consider the simple sections of pipework such as plane pipes and standard T's. It is this consideration of the simple parts of the emergency cooling system that is presented in this paper.

Preliminary calculations were carried out into the transient temperature distributions, and subsequent stress distributions of an 8 inch pipe in order to establish the acceptable water temperature. The 8 inch pipe was chosen as one of typical dimensions used in the initial design. The transient distributions were calculated by means of the finite element method incorporating an axisymmetric model, the temperatures in a triangular mesh, the nodes of which formed the quadrilateral isoparametric element mesh used in the stress calculations. As the problem of an infinite pipe can be reduced to one dimension which has analytical/numerical solutions, a comparison with the finite element was possible.

The boundary conditions of the problem were specified by the temperature of the pipe, the temperature of the water and the heat transfer coefficient between them. This final parameter was unknown and so a range was taken from 1000 to 10000 BTU/^{o}F.ft^{2}.hr. The heat transfer coefficient and water temperature were considered to be constant with respect to time. Once the water temperature was selected, with a heat transfer coefficient thought to be pessimistic, and yielding acceptable stress distributions, these boundary conditions were applied to the standard T's and similar temperature and stress distributions calculated.

Before assessing the effect of the thermal shock on the integrity of the pipe work, it is necessary to know the effects due to pressure loading. The circuit has three main pressure loads, an overpressure test, a maximum fault condition, and a normal operating condition. The critical length of a through axial crack in a plane pipe was calculated under these pressure loads. These revealed that the plane pipes were in a state of 'leak before break' under pressure loads and assumed fracture toughness of the material.

It was therefore possible, by assuming a crack growth law, to establish a maximum allowable through axial crack that would not grow to a critical size (under pressure conditions) over its lifetime (assuming 261 pressure cycles and 12 thermal shocks over 30 years). Partial semi-elliptical axial flaws on the inside surface of the pipe, where the high tensile thermal stresses occur, were then investigated. The length of the crack was fixed at the maximum allowable through crack defined above and its depth varied. Pressure and thermal loadings were considered separately, as it is assumed that pressure has dropped as a result of the accident before the thermal effects built up. The stress intensity factors, K, including edge and bulging effects, were calculated for the different geometries and compared with the reference K in ASME III for A533 steel. This was used as a guide for the toughness of the material. The thermal K values were calculated by the method of superposition, using the finite element thermal stress distributions and the known solution for a crack in an infinite plate loaded at its surfaces by a pair of wedge forces. These were compared with the K values obtained using the maximum surface thermal stress, assuming it was acting over the depth of the crack.

A flaw was therefore established that would not become critical, under the assumed loading conditions and fracture toughness of the material. This flaw was taken to be the 'reference' flaw at the end of its lifetime and subsequent integration of the crack growth law, yielded the dimensions of the 'reference' flaw. Assuming flaws of this size and greater can be detected during preservice inspection, the plain pipe should not fail during its lifetime due to thermal shock. The thermal fatigue driving stress was taken to be the maximum hoop stress occurring in the transient.

To carry out a similar analysis on the standard T's, they were considered to be plain pipes of thickness equal to the maximum thickness of the T. For the pressure stresses, a stress concentration factor of three was assumed and the subsequent K values for partial semi-elliptical defects of different aspect ratios calculated. The thermal calculations used the finite element thermal stress distributions of the T, corresponding to those through the section of maximum thickness. Reference flaws could then be established as in the case of the plain pipes.

2. Transient Thermal Calculations

2.1 Plain Pipes

The preliminary investigations into the transient distributions of the 8 inch pipe, of thickness 0.563 ins. and inner radius 4 ins., were carried out using an axisymmetric finite element idealisation. The mesh is shown in Figure 1, the dotted lines indicating the triangular elements used in the temperature calculations.

The boundary conditions were taken to be constant with respect to time and those considered are summarised in Table I. Typical transient temperature, and corresponding stress, distributions are given in Figure 2, the boundary conditions being a pipe temperature of 540^{o}F, a water temperature of 70^{o}F, and a heat transfer coefficient of 5000 BTU/^{o}F.ft^{2}.hr. between them. Also included is a pressure loading of 1000 psi, the normal operating pressure of the pipe work. The thermal transient distributions can be found by simply subtracting the analytical pressure stresses from the joint distribution. As expected, the high tensile thermal hoop and axial stresses occur at the inner surface of the pipe. The variation of the thermal hoop stress at the inner surface with time is plotted in Figure 3, for different values of heat transfer coefficient. The maximum thermal hoop stress is given in Figure 4 as a function of heat transfer coefficient for different values of water temperature.

It can be seen from these curves that the maximum thermal hoop stress occurs within five seconds of the start of the transient. They also appear to level off as the heat transfer coefficient is increased. It was therefore decided to restrict all further investigations to 5000 BTU/^{o}F.ft^{2}.hr. thought to be a pessimistic value of the heat transfer coefficient, and a water temperature of 70°F. It is the transient distributions of these boundary conditions that are used to assess the integrity of the 8 inch plain pipe.

2.2 Standard T's

Two standard T's were analysed, an 'equal' T, joining pipes both of radius 2.013 ins., and an 'unequal' T, joining pipes of radius 2.013 ins. and 4.25 ins. They were both approximated by an axisymmetric finite element idealisation of a pipe entering an infinite plate. The free end of the plate was constrained so that it was free to expand uniformly in the plane of the plate. This was found to model the real situation more realistically than a plate with a free end. A similar constraint was imposed at the free end of the pipe. The details of the two meshes are given in Figures 5 and 6. The boundary conditions were taken as a pipe temperature of 540°F, a water temperature of 70°F and a heat transfer coefficient of 5000 BTU/°F.ft^2.hr.

The transient temperature distributions of the free end of the pipe and plate, in both cases, agreed well with those of the appropriate dimensioned infinitely long pipe. However, the stress distributions did not match so well. This was attributed to the mesh not including sufficient length in the pipe, even though the temperature solution indicated that this was so. The temperature distributions through the section of maximum thickness are given in Figures 7 and 8. The corresponding maximum hoop stresses are shown in Figure 9. These are used in subsequent fracture analyses of the T's.

3. Application to Circuit Integrity

3.1 Fracture Toughness

The fracture toughness details for the proposed material were assumed to be as for A533 steel in ASME III (Subsection NA, Appendix G). This is taken as indicative of possible materials such as BS3602. The variation of K_{IR} with temperature is given by:

$$K_{IR} = 26.78 + 1.233e^{0.0145\,(T - RT_{NDT} + 160)} \qquad (1)$$

where the units of K_{IR} are ksi$\sqrt{\text{in}}$, RT_{NDT} is the nil ductility temperature, assumed as 0°F, and T is the temperature in °F. A maximum toughness of 250 ksi$\sqrt{\text{in}}$ is assumed. This reference toughness is established as the minimum of K_C, K_{IC} and K_{ID}.

For plain strain conditions to be considered dominant, the relationship

$$B \geqslant 2.5 \left(\frac{K}{\sigma_y}\right)^2 \qquad (2)$$

must be satisfied, where B is the thickness, K the stress intensity factor, and σ_y the yield stress. This means that there is a maximum fracture toughness level K_{IC} that can be measured for a particular material. This is given by:

$$K_{IC} \leq \sigma_y\,(0.4B)^{\frac{1}{2}} \qquad (3)$$

For the pipe geometries under consideration, the condition (2) is not satisfied. It is possible to run to COD concepts of general yielding fracture mechanics to obtain the fracture toughness of the material, at least in the case of through thickness defects. The COD values of the material are assumed to be in the range of 0.01 to 0.04 ins (100 to 150 ksi$\sqrt{\text{in}}$) depending on the temperature.

Burdekin and Dawes (1) give the relationship between the COD value, δ , a through crack of length 2c and an operating stress of σ in a pipe of radius R and thickness B as

$$\delta = \frac{8\sigma_y c}{E} \left[\log \sec \left(\frac{\pi \sigma}{2\sigma_y} \right) \right] M_B^2 \qquad (4)$$

where $M_B = \left[1 + \frac{1.61\ c^2}{R\delta} \right]^{\frac{1}{2}}$ is known as the bulging factor.

However, when partial defects, of length 2c and depth a, are considered, the stress intensity values are determined and compared with the reference toughness of A533. These were calculated by the method of superposition, given the stress distribution in an uncracked pipe, and the known solution for a crack in an infinite plate loaded at its surfaces by a pair of wedge forces. For full thickness cracks extending a depth of 'a' perpendicular to one of the plate edges, the analytical relation can be written, from Blanel et. al (2), as

$$K = 1.12 \quad 2\sqrt{\frac{a}{\pi}} \int_0^a \frac{\sigma(x)\ dx}{(a^2 - x^2)^{\frac{1}{2}}} \qquad (5)$$

The factor 1.12 in eq.(5) describes the influence of the stress free surface where the crack starts. Combining this equation, the edge effects of Maddox (3) and the bulging effect of the pipe, the K value can be written as

$$K = M_m M_B\ \sigma_1 \sqrt{a} \qquad (6)$$

where the values of M_m are given in Figure 10, and

$$\sigma_1 = \frac{2}{\sqrt{\pi}} \int_{R_i}^{R_i + a} \frac{\sigma(r)\ \ dr}{(a^2 - (r - R_i)^2)^{\frac{1}{2}}},$$

and R_i is the inner radius of the pipe. The σ_1 integral can be evaluated numerically by using an 8 point Chebyshev integration formula over half its range. If, as in the case of pressure loading, σ(r) is a constant, σ_1 , reduces to $\pi/2$.

Any fatigue growth is evaluated by means of the crack growth equation

$$\frac{da}{dN} = 7.7 \times 10^{-11}\ (\Delta K)^{3.83} \quad \text{ins/cycle} \qquad (7)$$

which is thought to be an upper bound for this material. However, it does not take into account the particular environment in question, and only really applies to the region where stable crack growth occurs. When the K values near their critical value, the growth of a crack becomes unstable and the above law need not necessarily apply.

3.2 Axial Flaws in Plain Pipework

3.2.1 Through Cracks

Before determining the effect of the thermal shock on the plain pipe, it is necessary to consider the pressure loading. Using eq.(4), the critical through crack lengths for the 8 inch pipe can be determined under the three pressure loads. These are summarised in Table II. Leak before break situations are assumed to exist in all cases as the crack length is several times the thickness of the pipe.

It is assumed that the pipework experiences 261 cycles of the maximum fault con-ition of 1210 psi over its lifetime of 30 years. The maximum allowable through crack under pressure loads can therefore be established by integrating the growth law, eq. (7), from the initial (3.1 ins. corresponding to the overpressure conditions) to the critical length. This yields a final crack length of approximately 3.5 ins., which is less than the critical length of 5.3 ins., under maximum fault conditions. The maximum allowable crack could then be defined as 3.1 ins. under pressure conditions.

However, when the thermal shock is included, it is necessary to ensure that any through flaw cannot grow to any of the critical lengths under pressure. The maximum allowable through crack is defined as the length of through crack that would just now grow to the assumed critical length of the overpressure test. This should ensure that no pressure considered would cause the pipe to fail. The thermal shock, of which the maximum surface hoop stress of 70 ksi is taken as the fatigue driving stress, is assumed to act 12 times over 30 years. Integration of the crack growth law, eq. (7), yields a maximum allowable through crack length (c) of 1.06 ins.

3.2.2 Partial Depth Flaws

Partial depth flaws can cause snap through failure and it is therefore necessary to investigate them. Since the high tensile thermal hoop stress occurs on the inside of the pipe, partial flaws in this region are more likely to be serious than those existing on the outside surface of the pipe. Two such types of defect were considered, one with an aspect ratio of $^a/2c = 0.01$, and the other of a length equal to the maximum allowable through crack. The depth ratio $^a/B$ was varied in both cases.

Most interest lay with the values of the stress intensity factors under thermal loadings during the transient. These were calculated by the method of superposition as described in Section 3.1, and also by assuming that the maximum surface stress acted throughout the depth of the crack. Figures 11, 12 and 13 show the variation with time and depth of flaw for both defects considered.

It is possible, by examination of these curves, to establish a 'safe' defect size under the transient thermal loads. A flaw of dimensions a = 0.2252 ($^a/B = 0.4$), c = 1.057 ins would be satisfactory, irrespective of how the thermal loads were defined, provided the toughness of the material at the temperatures of the shock was approximately 100 ksi$\sqrt{\text{in}}$. (Work by P.V.R.C. Ad Hoc Group (4) on toughness requirements state that, at about 60°F, the effective toughness of a SA106B pipe is 95 ksi$\sqrt{\text{in}}$). Any existing defect at the start of its life must not grow to this size during its lifetime. Integration of eq. (7) determined a 'reference' flaw of dimensions a = 0.18 ins., c = 0.85 ins., assuming that the aspect ratio remains constant.

Outer flaws of partial depth are then unimportant in the thermal context. For the thermal stress to build up over the depth of the crack, and so ensure large thermal K values, the crack depth must be at least into the tensile thermal stress zone on the inside of the pipe. A crack depth ratio of $^a/B = 0.4$ would ensure that this would never happen. Inside and outside flaws of 'reference' dimensions are clearly not critical under pressure loads.

3.3 Axial Flaws in Standard T's

3.3.1 Partial Depth Defects

The standard T's were modelled in the fracture analysis by pipes of a thickness equal to the maximum thickness of the T, and of a radius equal to that of the major one of the T. A stress concentration factor of 3 was assumed on the pressure stresses of the plain pipes. The thermal transient distributions were taken from the finite element solutions across the section of maximum thickness in the 2-D idealisation.

Partial depth flaws with aspect ratios of $^{a}/_{2c} = 0.1$, 0.3 and 0.5 were considered on the inside surface of the pipe. Larger ratios were not investigated as the area of high stress, thermal and pressure, is local to the crotch corner of the T. The stress intensity factors, assuming a constant maximum surface stress acting over the depth of the defect, are shown in Figures 14 and 15 for the two standard T's considered.

If a toughness of 100 ksi$\sqrt{\text{in}}$ is assumed at the temperatures of the thermal shock, defects can be defined that are critical under thermal, but not pressure, loads. Integration of eq.(7) determines a 'reference' flaw that must be detected to ensure that the approximated T does not fail. These are given in Table III.

It is worth noting that Sumpter and Turner (5) suggest a rule of thumb in connection with openings in pressure vessels. They say that if the crack size 'a' is greater than 0.2R, where R is the radius of the opening, then the effective crack size is 'a + 2R'. The minor radius of both T's considered here is 2.013 ins., and so the diameter of the inlet pipe does not have to be added to the depth of the flaw.

4. Discussion

The lack of details of the specific fracture toughness of many materials makes a realistic investigation into the integrity of emergency cooling systems difficult. However, by making assumptions from available data and by using the reference toughness of A533 steel as a guide, a reasonable analysis can be carried out into the simple parts of the pipework.

The investigation was confined to axial cracks. The critical lengths of through cracks in plane pipes under the three pressure loads were all greater than twice the thickness of the appropriate pipe, thus indicating a leak before break situation. Once the maximum allowable through crack was established, a 'reference' flaw of partial thickness was determined that was not critical under thermal loading and that would not grow through the thickness of the pipe during its lifetime. However, if it did, a leak situation would exist as its through crack length would be subcritical under any pressure loading. The approximation of the standard T's to plain pipes in order to establish a similar 'reference' flaw to that of the pipe, contains more uncertainties, but it does give a first approximation to the problem. If the 'reference' flaws are of a detectable size, as seems the case for the simple pipework considered, it is worthwhile proceeding with a more elaborate research programme.

It is important to note the main assumptions that were made in order that this first analysis could be carried out. They are:

1. The thermal stresses were evaluated by means of the finite element method and therefore contain uncertainties associated with the method.

2. The boundary conditions were assumed to be constant with respect to time.
3. The pressure and thermal loads were assumed not to act at the same time.
4. Fracture toughness properties were assumed as for A533.
5. The expressions for the partial depth defect do not include any plasticity effects.
6. The fatigue crack growth law does not take into account the environment in question and only strictly applies to the region of stable growth. During the growth of partial depth flaws, the aspect ratio is assumed not to vary.

5. Acknowledgements

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6. Appendix: Material Properties of BS3602 - HFS27, assumed for this study

Yield Stress (0.2% Proof Stress)

Temperature ^{o}F	70	100	200	300	400	500
Yield Stress ksi	35.8	35.4	31.6	28.7	27.3	25.5

Minimum U.T.S. = 60 ksi

Youngs Modulus E = 26×10^{6} psi

Poissons Ratio $\nu = 0.3$

Density $\rho = 0.2836$ lb/in^{3}

Coefficient of Thermal Expansion $\alpha = 7.1 \times 10^{-6}$ $^{o}F^{-1}$

Thermal Conductivity k = 0.000674 BTU/in.sec.^{o}F

Specific Heat Cp = 0.13375 BTU/lb.^{o}F.

Pressure Loads

Overpressure Test	1650 psi
Maximum Fault Condition	1210 psi
Normal Operating Condition	1000 psi

7. References

(1) BURDEKIN F.M., DAWES M.G., "Practical Use of Linear Elastic and General Yielding Fracture Mechanics with Particular Reference to Pressure Vessels", Conference on Practical Application of Fracture Mechanics to Pressure Vessel Technology, I. Mech. E., London, May 1971.

(2) MADDOX S.J., "An Analysis of Fatigue Cracks in Fillet Welded Joints", Welding Institute Research Report E/49/72, January 1973.

(3) BLANEL J.G., MALTHOFF J.F., STAHN D., "Model Experiments for Thermal Shock Fracture Behaviour", J. Engng. Mats. and Technol., October 1974.

(4) P.V.R.C. Ad Hoc Group on Toughness Requirements, "Recommendations on Toughness Requirements for Ferratic Materials", WRC Bulletin 175, August 1972.

(5) SUMPTER J.D.G., TURNER C.E., "Fracture Analysis in Areas of High Nominal Strain", 2nd International Conference on Pressure Vessel Technology, San Antonio, Texas, October 1973.

Table I: 8 inch Pipe Boundary Conditions

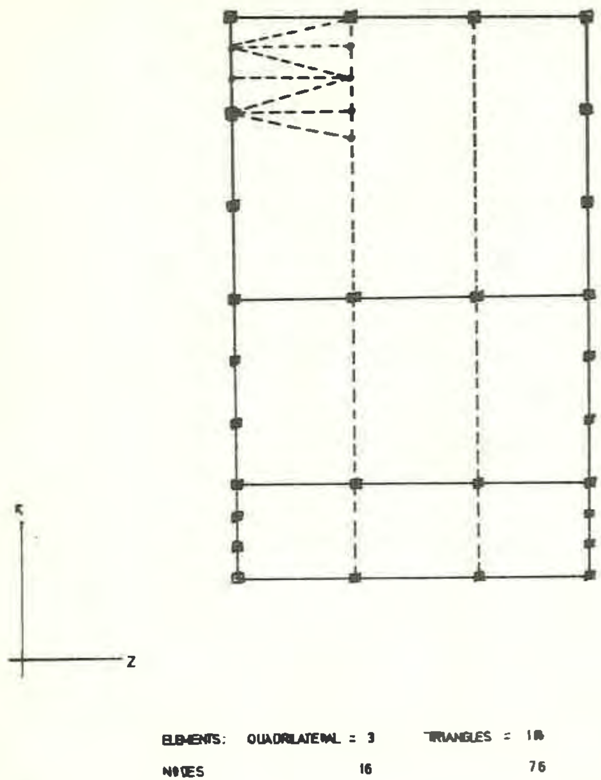
Pipe Temperature ^{o}F	Water Temperature ^{o}F	H.T.C. $BTU/^{o}F.hr.ft^2$
540	390	500
540	390	1000
540	390	5000
540	100	1000
540	100	2000
540	100	5000
540	100	10000
540	70	1000
540	70	2000
540	70	5000
540	70	10000

Table II: 8 inch Pipe Pressure Details

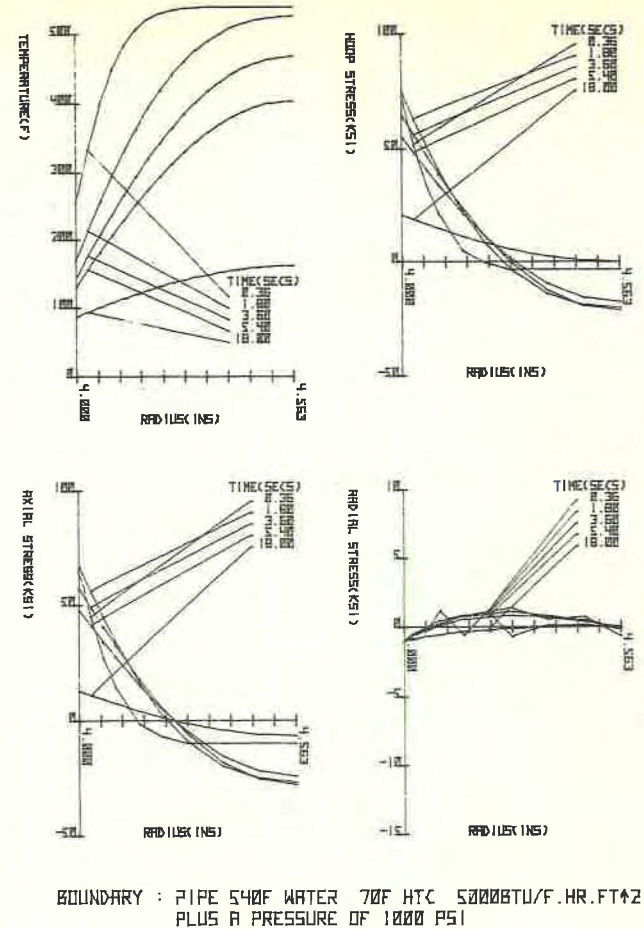
Pressure psi	Hoop Stress psi	Temperature ^{o}F	Yield Stress psi	C.O.D. ins	a_c ins
1650	11723	122	34500	0.01	3.1
1210	8597	500	25500	0.04	5.3
1000	7105	500	25500	0.04	6.1

Table III: Standard T Details

Type of T	Inner Radius ins	Max. Thickness B (ins)	K = 100 ksi in $^a/_{2c}$ = 0.1 $(^a/_B)$ crit.	$(^a/_B)$ ref.
Unequal	4.25	0.64	0.3	0.248
Equal	2.013	0.322	0.5	0.316



1. Idealisation of 8 inch pipe



2. 8 inch Pipe: Transient Temperature and Stress Distributions

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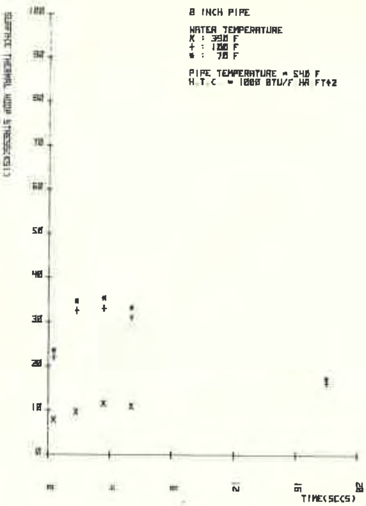
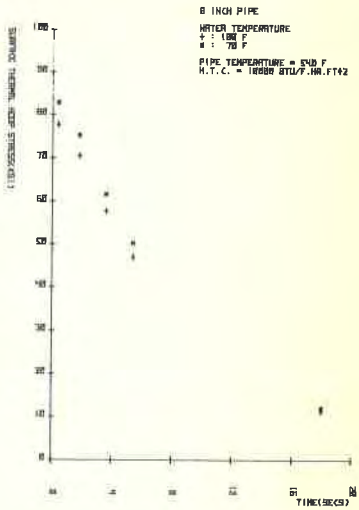


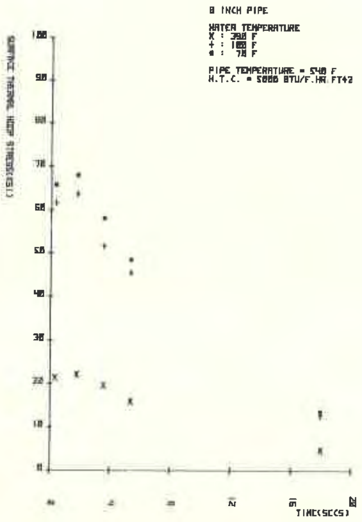


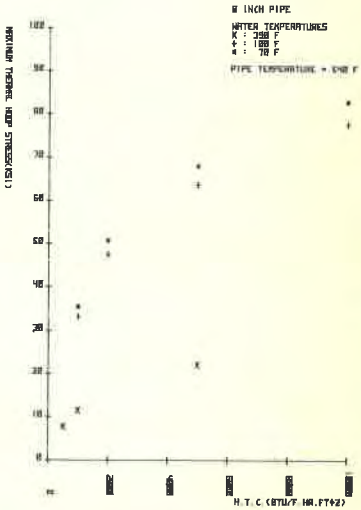
3. 8 inch Pipe: Surface Thermal Hoop Stress vs. Time for

(a) H.T.C. = 1000 BTU/^{o}F.ft^{2}.hr.

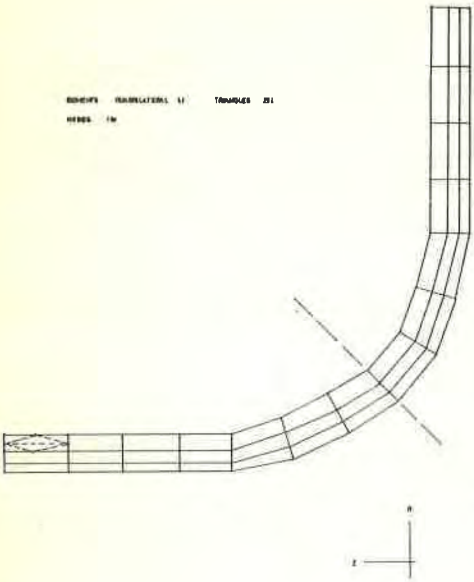
(b) H.T.C. = 5000 BTU/^{o}F.ft^{2}.hr.

(c) H.T.C. = 10000 BTU/^{o}F.ft^{2}.hr.

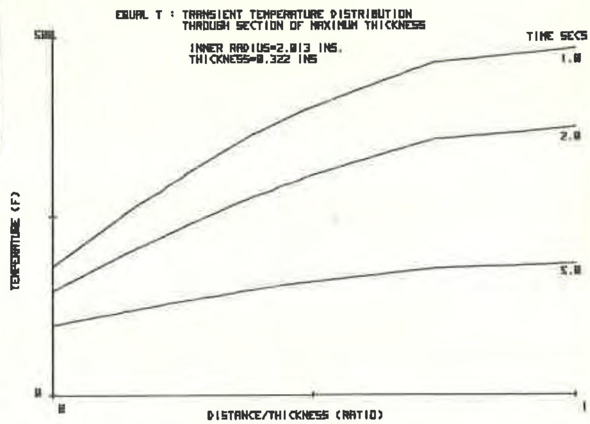




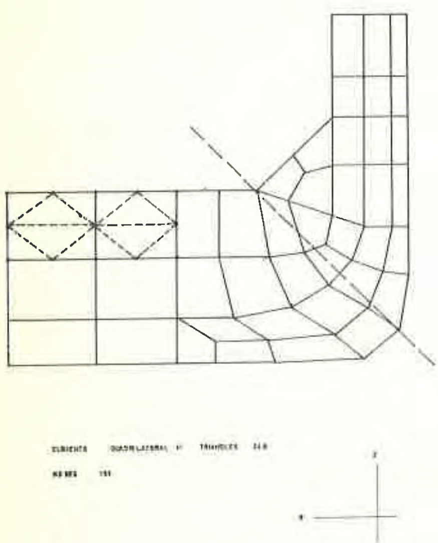
4. 8 inch Pipe: Maximum Thermal Hoop Stress vs. H.T.C.



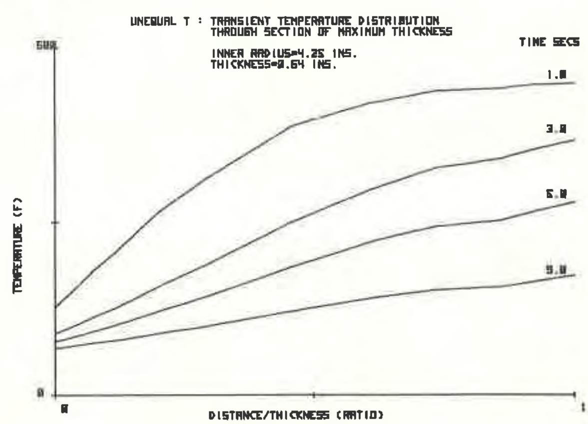
5. Idealisation of Equal T



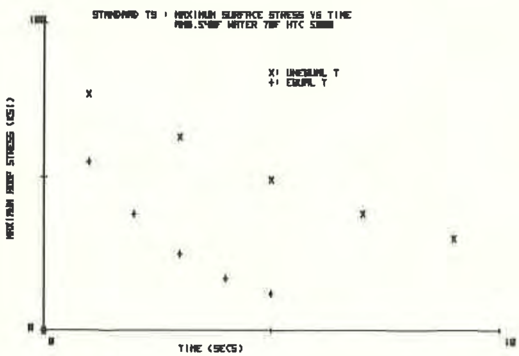
7. Equal T: Transient Temperature Distributions

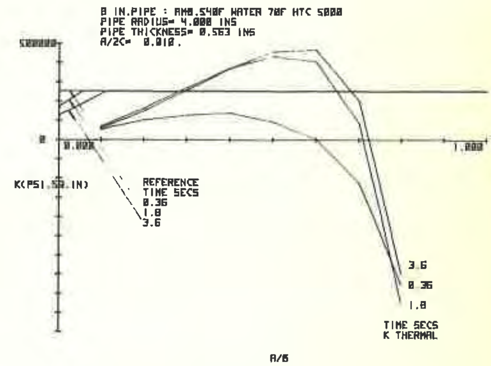


Idealisation of Unequal T



8. Unequal T: Transient Temperature Distributions



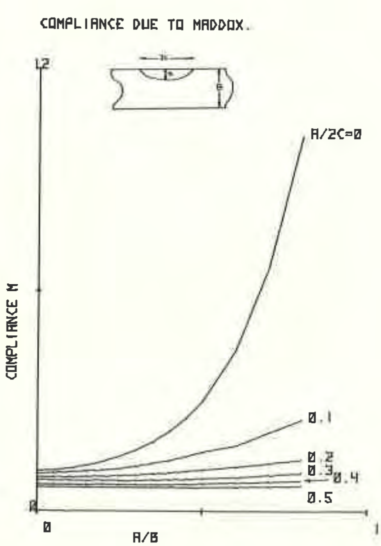


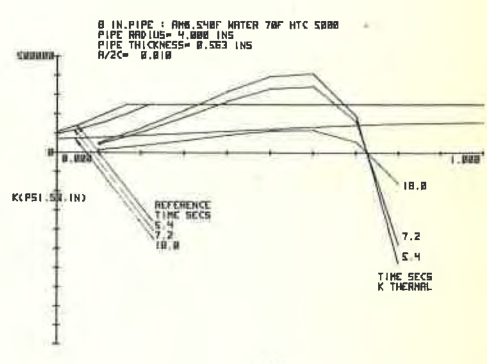
9. Standard T's: Maximum Surface Stress vs. Time

11. 8 inch Pipe: Thermal K values of defect $^{a}/_{2c} = 0.01$

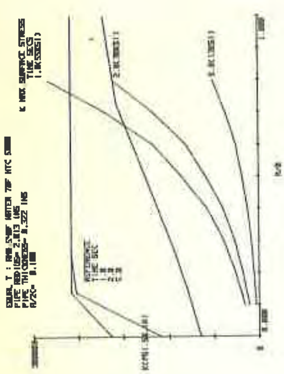
(a) Times 0.36, 1.8, 3.6 seconds

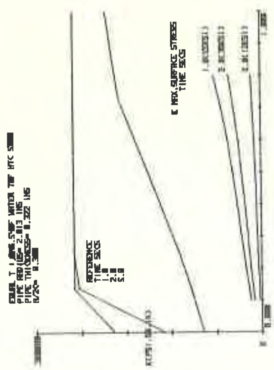
(b) Times 5.4, 7.2, 18.0 seconds

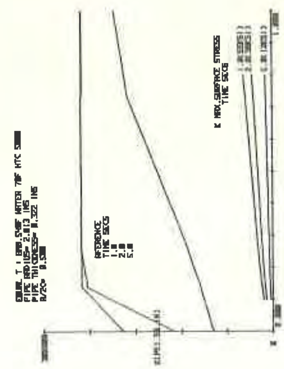




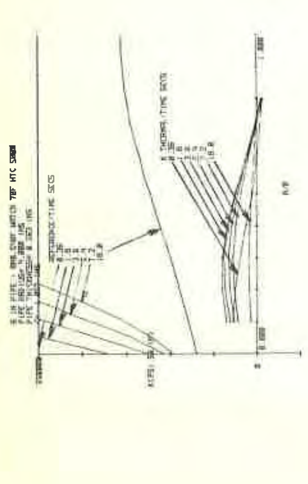
10. Compliance of Semi-Elliptical Flaws



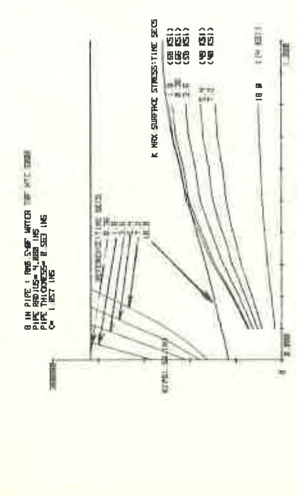




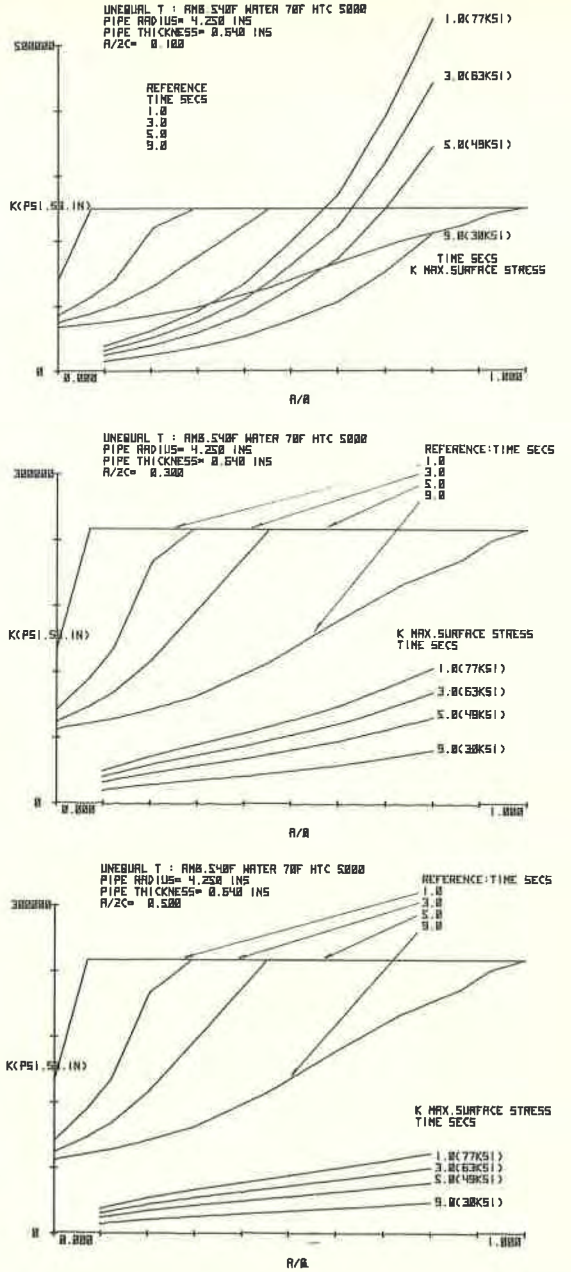
14. Equal T: Thermal K Values (Maximum Surface Stress) of Defect
   (a) $^{a}/_{2c} = 0.1$
   (b) $^{a}/_{2c} = 0.3$
   (c) $^{a}/_{2c} = 0.5$



12. 8 inch Pipe: Thermal K Values of Defect of Length = Maximum Allowable Through Crack



13. 8 inch Pipe: Thermal K Values (Maximum Surface Stress) of Defect of Length = Maximum Allowable Through Crack



15. Unequal T: Thermal K Values (Maximum Surface Stress) of Defect

(a) $a/2c = 0.1$

(b) $a/2c = 0.3$

(c) $a/2c = 0.5$