MAIN STEAM PIPE FAILURES

BY

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Two main steam pipe failures occurred in one of H.M. ships during the past year. Investigation into these failures emphasized important considerations for all who are concerned with the design, fabrication, installation and maintenance of high temperature pipe systems.

Before reviewing the failures, an appreciation of the design criteria for the Class would be of interest.

Class Design

In the early stages of design, following previous Admiralty practice, a hoop stress for the pipe systems of 6,000 lb/sq in. was adopted. As this Class of ship was the first British marine application designed to operate with a steam temperature of 850 degrees F., a recommendation to raise the hoop stress to 10,000 lb/sq in. was considered. This recommendation was made with the object of :—

(*a*) Reducing bending moments at flanged joints to minimize the possibility of joint leakage

(*b*) Reducing thrusts on terminal machinery

(*c*) Saving in weight.

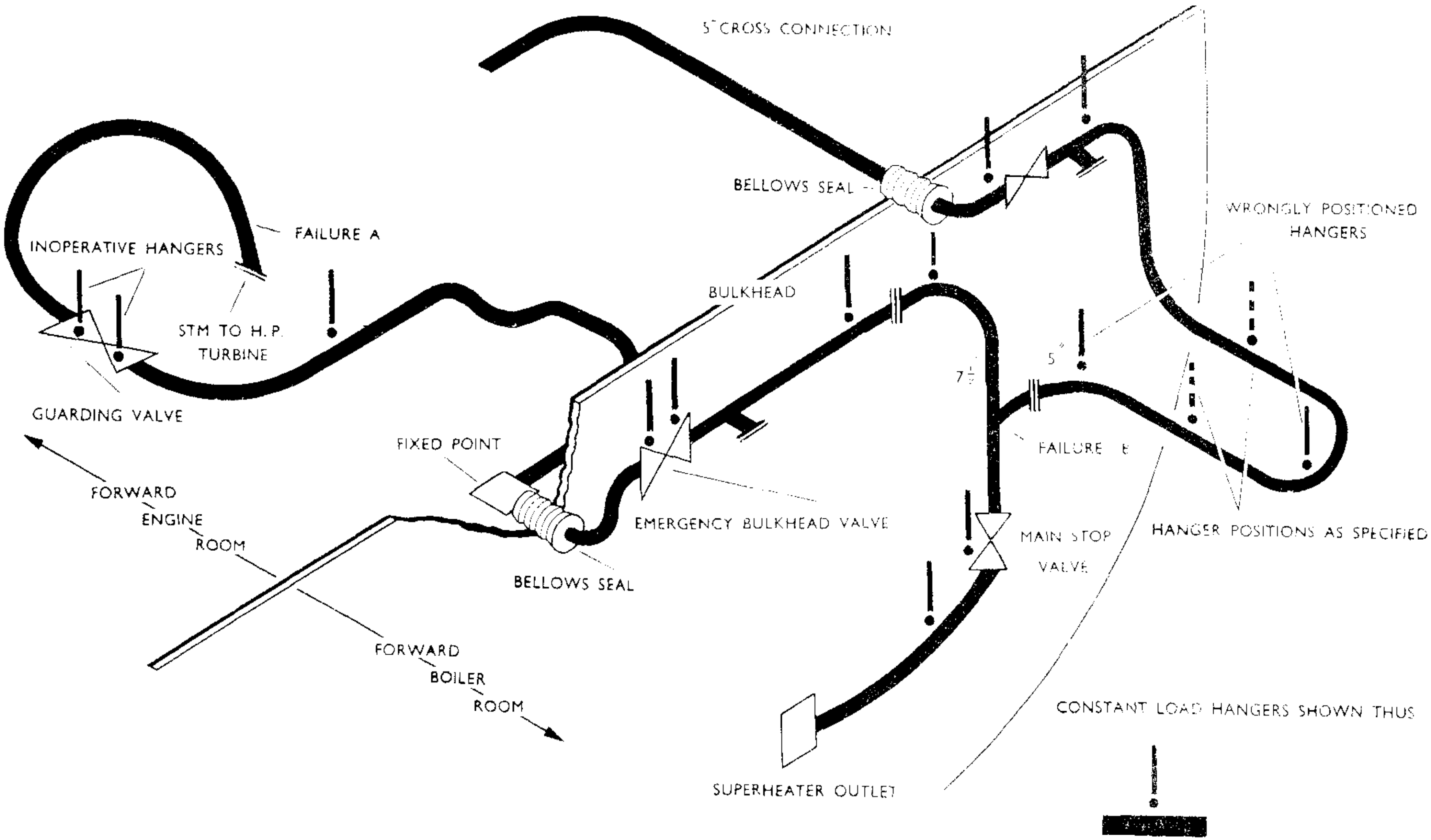

FIG. 1—DIAGRAMMATIC LAYOUT OF STEAM SYSTEM

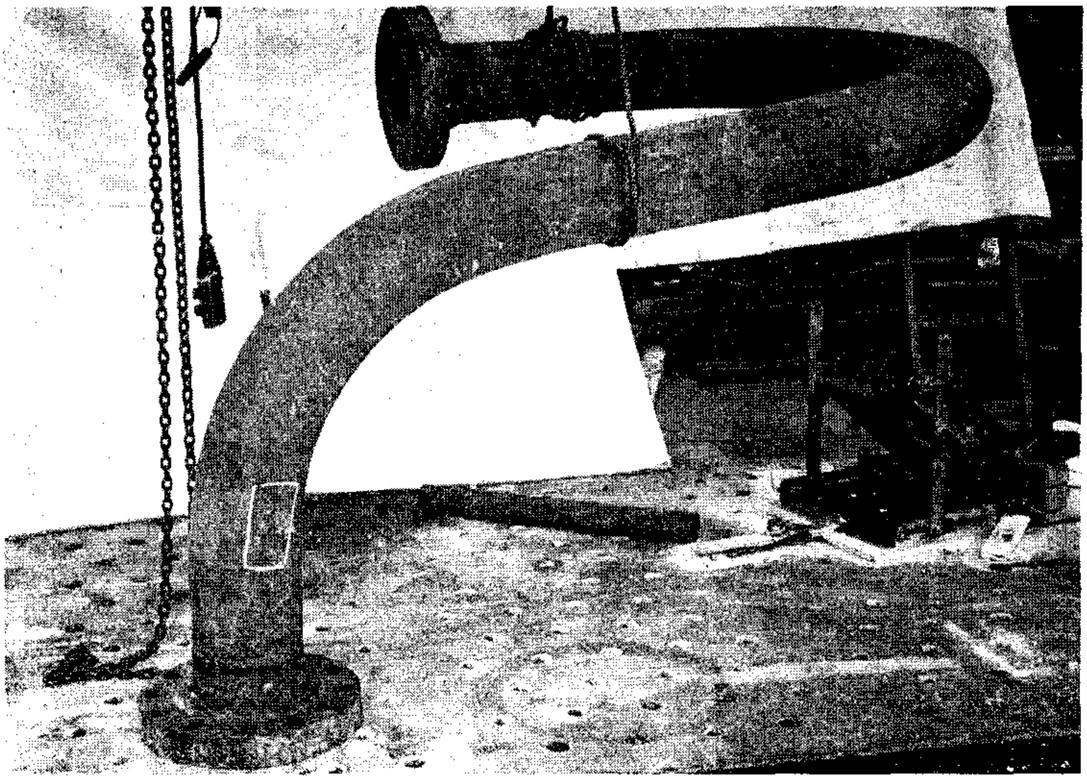

FIG. 2—DEFECTIVE STEAM PIPE SHOWING LOCATION OF CRACKING

At this particular time, thought was being given to the use of chrome-molybdenum steel for high steam temperatures in view of its creep resistance properties. Some failures with carbon-molybdenum steels in power stations had occurred but were still the subject of investigation. In order to gain experience with various types of piping, it was decided that the criteria for systems in the ships of the Class should be as follows :—

Four ships—carbon molybdenum, hoop stress 6,000 lb/sq in.

Two ships—carbon molybdenum, hoop stress, 10,000 lb/sq in.

Two ships—chrome molybdenum, hoop stress, 10,000 lb/sq in.

The four latter ships were regarded as experimental from the pipe system material and flexibility aspects, although their machinery layouts and consequently their piping layouts are not identical in all respects.

The Failures and Related Power Station Experiences

The two pipe failures occurred in one of the ships fitted with 10,000 lb/sq in. hoop stress design carbon-molybdenum piping and appear analogous to those experienced in power stations.

Subsequent to the building of this Class of ship the British Electrical and Allied Industries Research Association's report on the cracking in service of 0·5 per cent molybdenum steam pipes became available.

The report covers some 49 failures of carbon-molybdenum pipes noting that many occurred at stations after changing to a two-shift operation. Steam temperatures in the stations ranged from 850 to 975 degrees F. while steaming time to failure varied from 3,000 to 40,000 hours.

By the time this report was finalized the Admiralty had decided to specify chrome-molybdenum steel for future designs with steam conditions of 850 degrees F. and above.

The two ship failures are surveyed separately, and treated as 'Failure A' in the for'd engine room and 'Failure B' in the for'd boiler room. The system layout is shown diagrammatically in FIG. 1.

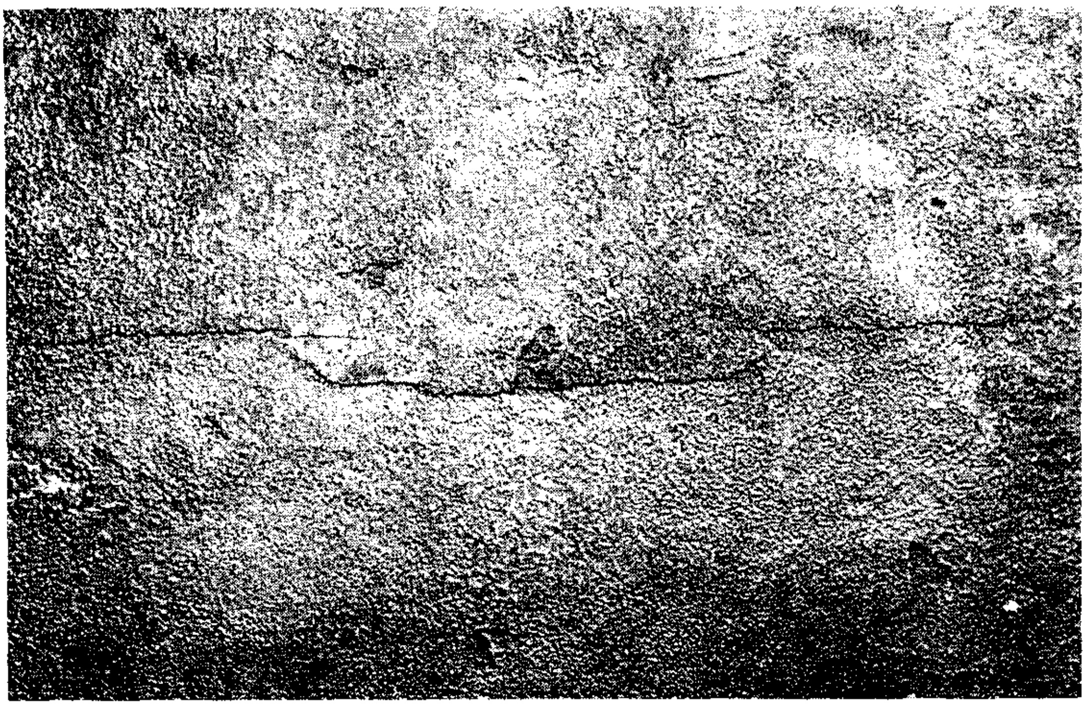

FIG. 3—APPEARANCE OF THE CRACKING ON THE OUTSIDE OF THE PIPE

Failure A

Pipe—$7\frac{1}{2}$ in. bore. 0·244 in. thick
Working conditions—650 lb/sq in., 850 degrees F.
Total steaming time—7,800 hours.

Failure occurred on the neutral axis at the commencement of a bend of 24 in. radius on the main steam pipe immediately above the flange adjoining the H.P. turbine nozzle control valve chest. (See FIGS. 2 and 3).

General Considerations

The following relevant facts were established during investigation into the failure :—

(i) The designed allowance of $\frac{7}{16}$ in. cold pull-up in the fore and aft direction no longer existed

(ii) The double constant-load pipe hangers at the turbine guarding valve were inoperative

(iii) Bulkhead distortion existed in the vicinity of the steam strainer support which constitutes the fixed point of the system.

While bulkhead welding distortion did occur during building there was evidence that further movement had taken place before the failure since the emergency bulkhead valve spindle, geared through the bulkhead, had seized and was only freed by realignment. This movement could have contributed to (i) above. In this connection it should be noted that while it is specified that pipes be erected with cold pull-up the pipes are in fact designed to take full thermal expansion. The inoperative constant load pipe hangers supporting the guarding valve were, therefore, considered to be the main contributory factor causing failure.

Inspection revealed that the hangers were inoperative because the spring barrels at the rear of the spring retaining caps were internally coated with paint and deposits of cork chippings : this condition indicated that the hangers had been in this state since the ship was built. The hanger rods are connected

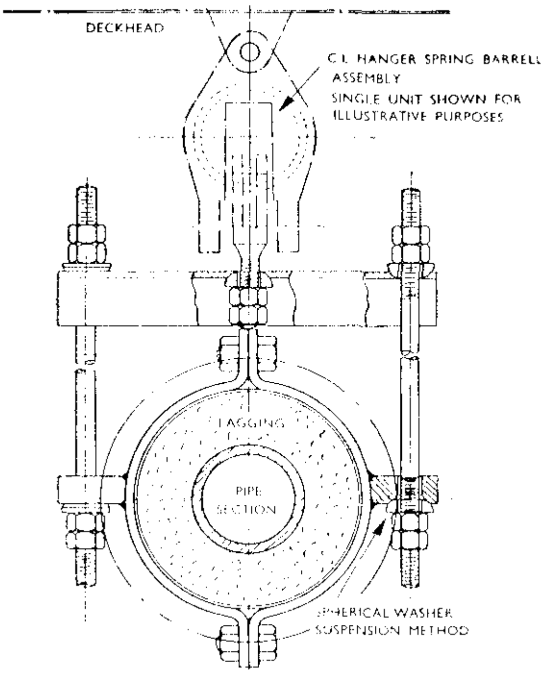

FIG. 4—CONSTANT-LOAD HANGER

by a yoke pivoting on spherical washers shown in FIG. 4. The inoperative state of the hangers resulted in the weight of the turbine guarding valve (11 cwt.) being imposed on the pipe system in the hot condition.

The original stress calculation taking into account full pipe thermal expansion and turbine movements only, showed that axial stress due to bending moment at the point of failure was 5,600 lb/sq in.

Assuming the system in the hot condition, with the additional weight of the guarding valve also imposed on the system, stress calculation showed the axial stress due to total bending moments at the point of failure to be 10,886 lb/sq in.

The transverse bending stress factor for the pipe bend (wall thickness—0·244 in., bend radius 24 in.) was approximately 3·15 and resulted in a transverse stress at the point of failure of 34,300 lb/sq in. whereas a reasonable maximum figure for this material is considered to be 28,000 lb/sq in. (For 6,000 lb/sq in. hoop stress ships, wall thickness 0·447 in. bend radius 30 in., the transverse bending stress factor was 1·61.)

No account was taken in these calculations of the hoop stress due to internal pressure, minimum value 10,000 lb/sq in., and the additive stress due to the known ovality of the pipe section, which has been estimated to be in the order of 16,500 lb/sq in. Some 13 per cent reduction in wall thickness will also result due to bending.

While certain assumptions were made in these calculations the stresses quoted above indicate that the pipe could have been subjected to excessively high stress concentrations.

Metallurgical Considerations

Metallurgical examination established the following :—

(i) Failure was due to a creep crack originating from the outside of the pipe wall

(ii) There were no indications of surface damage to the pipe

(iii) Chemical analysis of the alloy steel conformed to that specified

(iv) There was a variation in the tensile strength of the pipe near the area of failure, i.e. 42 tons/sq in. with Diamond hardness V.P.N./30 of 165–171 compared with an average figure of 35 tons/sq in. and hardness 132–165 for other portions of the pipe.

In view of (ii) above, creep failure can be attributed to overstressing. The condition at (iv) is indicative of irregular heat treatment. It should also be noted there were indications of slight surface decarburization, suggesting that local torch heating may have been applied during the fitting of the pipe. (See FIG. 5).

Failure B

Fabricated branch pipe, 5 in. bore, $\frac{3}{16}$ in. thick, adjoining a pipe of $7\frac{1}{2}$ in. bore, 0·244 in. thick.

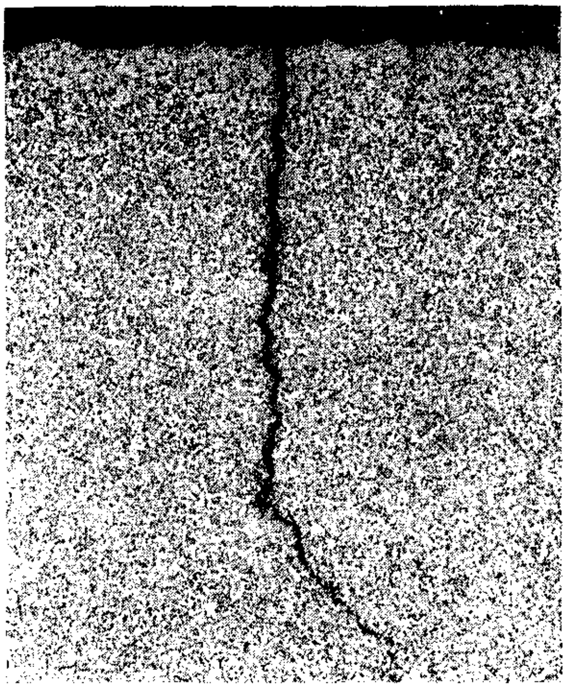

FIG. 5—MICROGRAPH SHOWING ONE OF THE MAIN CRACKS AND THE SURFACE DECARBURATION OF THE PIPE

Working conditions—650 lb/sq in., 850 degrees F.

Total steaming time—8,400 hours.

Failure occurred on the underside of the branch, in a circumferential direction, at the toe of the fillet weld securing the reinforcing strap where the branch enters the main pipe. FIG. 1 (boiler room) shows the system layout and FIGS. 6 and 7 show the detail.

General Considerations

The following relevant facts were established :—

(i) Failure occurred with steam on the 5 in. cross-connection and the forward boiler cold up to the main stop valve. The ship was steaming on the after boiler with systems de-unitized which had been a frequent condition of steaming

(ii) All hangers on the boiler room pipe arrangement were in a poor state of maintenance and their efficient operation was doubtful

(iii) A constant load hanger was sited on the bend immediately adjoining the 5 in. branch and adjusted to allow for upward expansion.

The condition of operation referred to at (i) is the most adverse from a stress aspect and was considered at the design stage. Calculation showed that the condition was acceptable with the designed support arrangements.

The siting and adjustment of the hanger quoted at (iii) above in this ship was not as in the Guidance Instructions issued at the time of building. The 'as fitted' and 'as specified' positions of the hangers for the cross-connection are illustrated in FIG. 1.

Metallurgical Considerations

Metallurgical examination established the following :—

(i) Failure was due to a creep crack originating from a corrosion fatigue fissure on the outside of the pipe at the toe of the fillet weld

(ii) Corrosion fatigue fissures existed on the external surface in the neighbourhood of the main crack

(iii) Chemical analysis of the alloy steel conformed to that specified.

The corrosion fatigue phenomena could have been produced by a leak from the joint, or saturation of lagging caused by a leak from an overhead system.

Incorrect hanger restraint is considered to have induced a high stress in the branch. This, coupled with the presence of corrosive conditions referred to at (i), is considered to have initiated corrosion fatigue cracking, leading ultimately to failure by creep cracking.

Conclusions

The investigations established that the failures were due to creep cracking resulting from excessive stress concentrations and are generally in line with the carbon-molybdenum steam pipe failures experienced in power stations.

Correct fabrication, installation and maintenance of high temperature steam

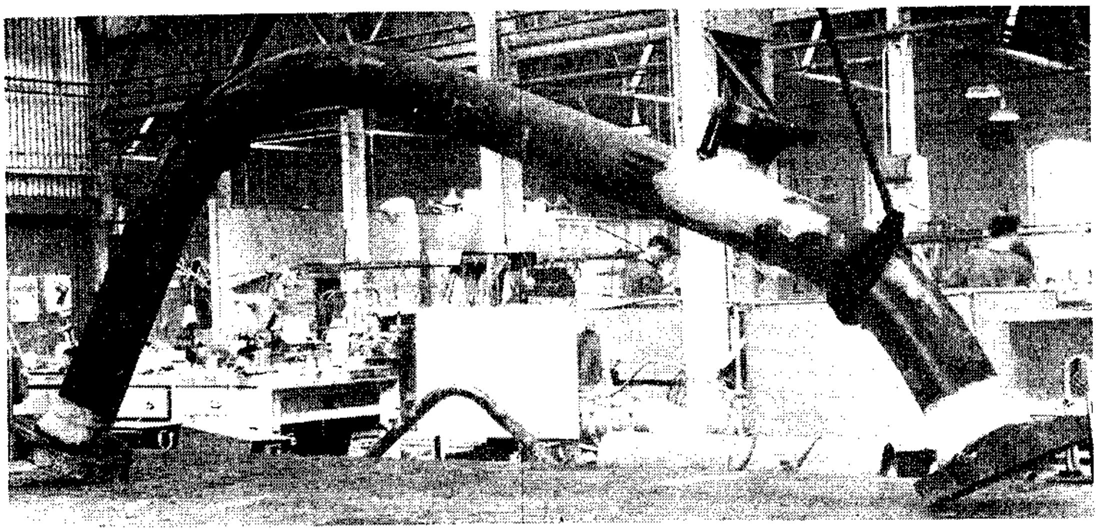

FIG. 6—APPEARANCE OF DEFECTIVE PIPE. CRACK IS ON THE BRANCH PIPE

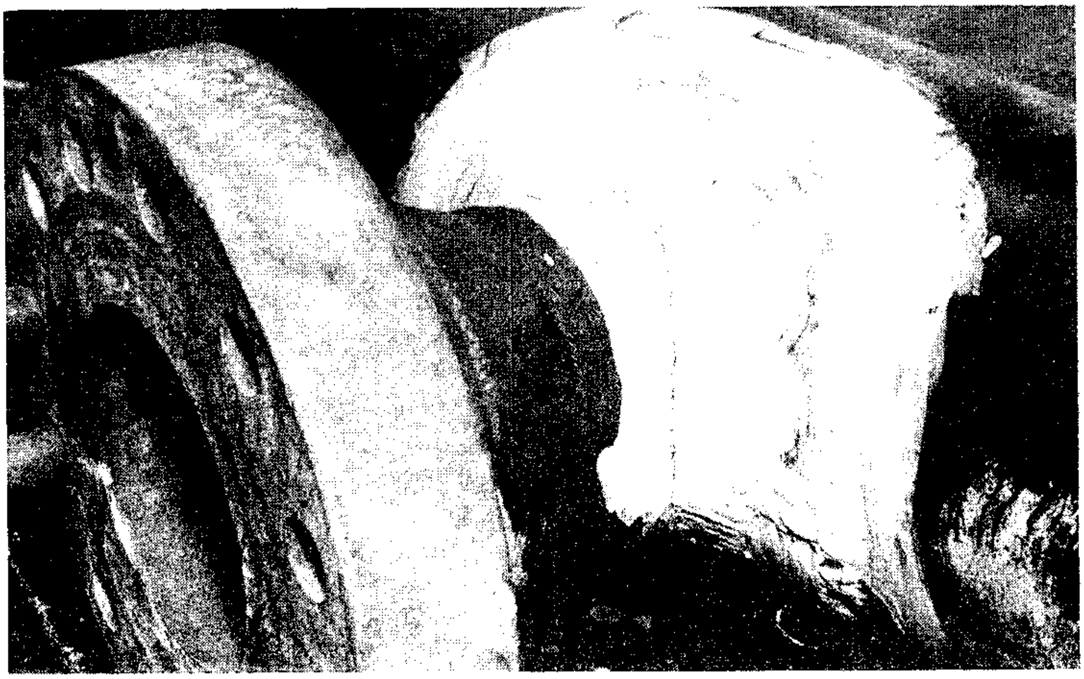

FIG. 7—CLOSE-UP OF BRANCH PIPE SHOWING CRACK AT THE TOE OF THE REINFORCING STRAP FILLET WELD

pipe systems is of the utmost importance. A most essential point in maintenance is to ensure that pipe hangers are fully operative.

Nothing has been proved wrong with the high hoop stress reasoning, but events have shown that where additional forces may be imposed under installed conditions the designer must seriously take into consideration the margin remaining between calculated and permissible bending stresses. More important still is the related transverse or circumferential bending stress at a pipe bend which increases considerably with any reduction in wall thickness.

The location and correct support of large valves and fittings in the pipe system must also be thoroughly investigated. Ideally, these weighty items should be positioned at the anchor points of the system.