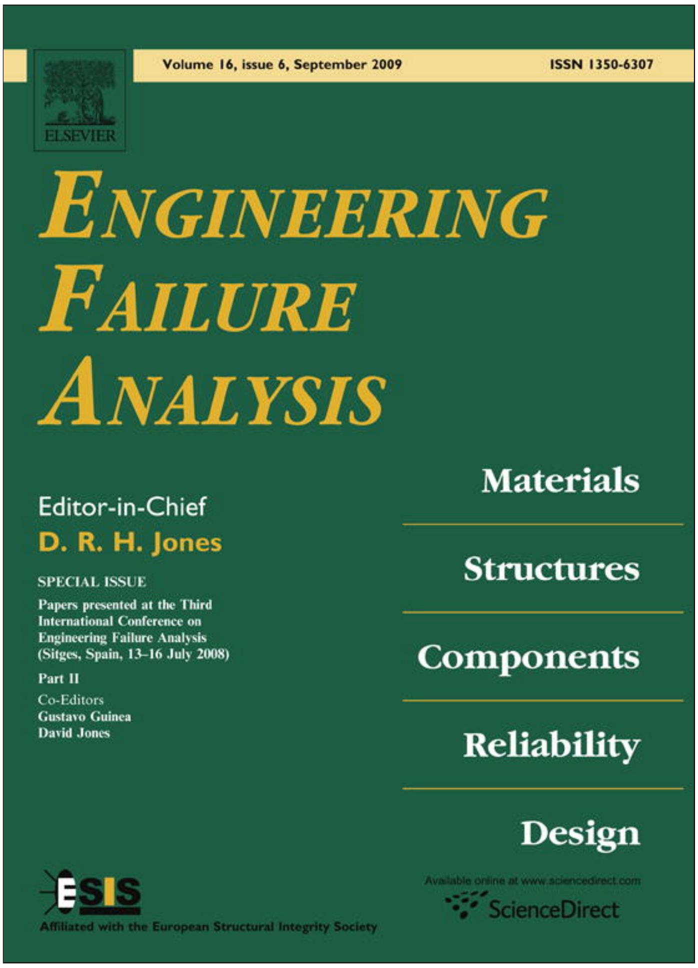
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# Common root causes of recent failures of flanges in pressure vessels subjected to dynamic loads

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ABSTRACT

Present rules in fabrication codes are aimed to reduce the probability of fatigue cracks in flanges welded to pressure vessels subjected to cyclic pressure or vibrations. Yet, several leaks and ruptures have recently occurred at flanges in pressure vessels and pipes. A review of three cases is presented, which involved five failures; their common root causes are discussed, and the influences of manufacture and operation conditions on crack initiation and propagation mechanisms are highlighted. Some cracks initiate from the outer surface, but many cracks initiate in the outer half of the thickness of the reinforcement, from very difficult to avoid weld defects. Ultrasonic testing, with an adequate procedure, can be reliably used to detect these defects before they become leaks.

Common aspects of these failures are crack initiation in weld metal and heat affected zone of welds, all related to an inadequate design of the reinforcement and poor execution of the welding procedure. Weld inadequacy is in one case a result of an increase in thickness as an attempt to increase safety; which also increased cyclic stresses due to excessive weight of the vessel. Fillet welded reinforcements induced rigidity inconsistencies within the flange joint, and concentrated deformations in the failed regions. In another case, pre-operational hydrotest could have been detrimental.



## 1. Introduction

Fatigue cracks in flanges welded to pressure vessels subjected to dynamic loads due to cyclic pressure or vibrations transmitted through the vessel and pipe bodies have been very frequent in the past [1,2]. Present rules in fabrication codes are aimed to reduce their probability. Yet, it has been found that several leaks and ruptures have occurred in pressures vessels in the gas and petrochemical industries [3]. The derived costs of some of these failures have been large, for what it is interesting to determine the influence of constructive and operation aspects, and possible mitigation alternatives. In this paper, a review of some of these cases is presented and their common root causes highlighted and discussed. The analysis is focused in the five following failures, all related to a crack in the reinforcement saddle of a pipe flange welded to the vessel:

- Case 1: Two discharge vessels at an alternating-compressor in a petrochemical plant.
- Case 2: 12″ collector pipe in a natural gas turbine-compressor plant.
- Case 3: Suction vessels in two natural gas alternating-compressor plants.

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These cases were studied using the usual techniques for analysis of mechanical failures:

- Analysis of operation and construction conditions.
- Visual and NDT inspections.
- Fractographic analyses.
- Mechanical and chemical analyses of plate reinforcement and saddle materials.
- Microstructural characterization.

in order to get conclusions related to:

- Characterization of crack propagation into vessel material.
- Influence of manufacture on crack initiation and propagation mechanisms.
- Influence of operation conditions, and possible mitigation measures.

## 2. Discharge vessels at an alternating-compressor in a petrochemical plant

Original and replacement vessels failed the same way. They were located in a polymerization line, subjected to the characteristic conditions of pressure and vibrations of an alternating-compressor [4]. The original vessel was designed in 1979 according to ASME PVBC code, section VIII Div. 1[5], was built in 1991, and failed four years later. Operation conditions are: temperature 110 °C, pressure 5 bar, and gas flow 4000/4500 Kg/h. The inner fluid is Ethylene plus a small quantity of solvent and serums coming from the low pressure recycle. The compressor has a rotation frequency of 375 rpm. The vessels are supported by means of elastic supports located in its inner part (grey arrow in Fig. 1), which were never subjected to non-destructive tests.

Material from saddle and reinforcement plate are common C–Mn ferritic pearlitic structural steels, plate material is 0.10 C and 0.66 Mn, while saddle material is 0.14 C and 1.33 Mn. Sulfur and phosphorus levels are average 0.014–0.022%, less than the 0.03% upper limit required for these steels. Material micro-hardness results for base and weld materials close to the area of the crack is around 20 HRc, while 30 HRc hardness values were found in some spots in the saddle HAZ. Hard spots were confirmed by course grained martensitic microstructures in HAZ.

Leaks developed where indicated by white arrow in Fig. 1, at the end of a longitudinal reinforcement of the pipe inlet. Pipe and reinforcement are 6.7 mm thick, saddle and vessel are 12.8 mm (design thickness of the vessel shell was 9.5 mm). Excess heat input caused undercuts in HAZ of the thinner plate. Brittle HAZ developed from too high cooling rates in saddle material. Discontinuous fillet welds were specified for the vertical reinforcement of the pipe inlet, these welds created high stress raisers at the ends of weld, and especially at the end of the reinforcement plate.

The crack initiated at the root of the fillet weld, and grew parallel to the surface of the vessel wall, on the outer surface of the saddle, see Fig. 2a–c. The crack path followed the weakest section in the fillet weld, near the end of the reinforcement of

**Fig. 1.** Case 1: Elastics supports of discharge vessels.

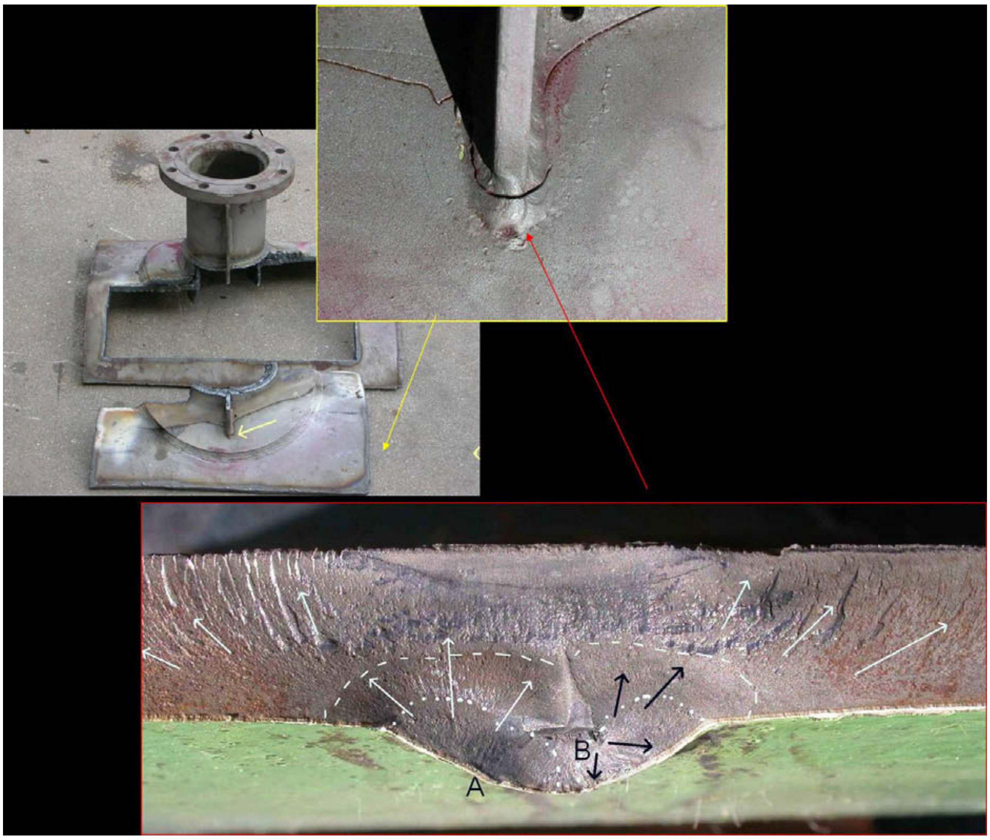


**Fig. 2.** The cracked area.

the pipe inlet, in the course grained HAZ of the saddle plate. The white arrows in Fig. 2c schematically indicate the direction of crack advance. The fracture surface shows the typical flat morphology indicative of propagation by mechanical fatigue. Very delicate and spaced beach marks are observed, indicating that the amplitude of the cyclic stresses were mostly constant during the whole crack propagation process.

Crack propagation was due to small amplitude stress cycles, during a very large number of cycles. A change in the roughness of the fracture surface is apparent in the upper part of the crack in Fig. 2c, with a river pattern marks parallel to the crack growth direction, indicative or faster crack growth. The fracture surface grows inclined with respect to the thickness, along the perimeter of the vessel, with a butterfly wing shape that completely penetrates the thickness of the saddle and partially penetrates the vessel wall. The deepest point of the crack is 92.3% of the vessel thickness, at the position of the weld root. The remaining ligament was of only 1 mm. A gas leak was about to occur if no remedial action were taken.

The spring-loaded supports located at the bottom of the vessel (grey arrow in Fig. 1) include frictional dampeners, which showed indications of excessive wear. All signs of malfunction are consistent with the excessive weight of the vessel, due to being much thicker than designed.

After the first failure, the plate reinforced flange was replaced by a weldolet – type welded flange, see Fig. 3. However, this new flange connection was also built with a lengthwise reinforcement welded with discontinuous fillet beads. A new failure occurred, this time in a shorter time, at the same position, see insert in Fig. 3. As expected, a finite element model of stresses due to dead weight and internal pressure (Fig. 4) shows that the end of the weld between reinforcement plate and saddle is the highest stressed region.

The failure of this partial redesign was due to the incomplete understanding of the stresses that were being generated at the initiation site. The stress concentration at the end of the pipe reinforcement was left unresolved. In this case crack growth occurred solely through the vessel wall, which led to an even reduced fatigue live. This study reveals that:

- The initiation of the crack that gave origin to the failure was located in the heat affected zone of the plate to saddle fillet weld, and is related to an inadequate design of the reinforcement and of the welding procedure. High hardness of HAZ and lamination defects in saddle material contributed to generate weak spots for crack initiation.
- Weld inadequacy is mostly a result of the large difference in thickness between the actual vessel and saddle (12.8 mm) and the design thickness (9.5 mm). Subsequent fatigue propagation of the cracks is related to a large number of cyclic loads, most probably due to vibrations coming from the compressor.
- Fatigue stresses were larger than expected also because the excessive weight of the vessel was not properly balanced by the elastic foundation, designed for a thinner, 25% lighter vessel.

**Fig. 3.** Welded flange in the second vessel, replacement of the failed vessel.

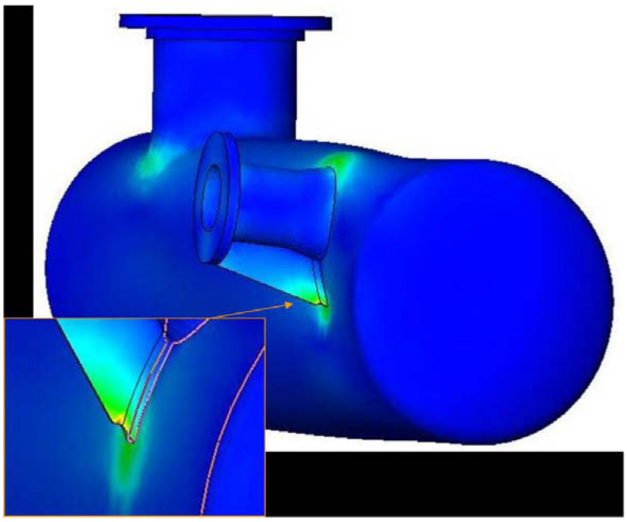

**Fig. 4.** Finite element model of stresses due to dead weight and internal pressure.

- The discontinuous fillet welded longitudinal reinforcement plate increases bending rigidity of the flange joint, and concentrates deformations in the failed region. It is not a mechanically very efficient design.
- Changes in elastic supports and NDT verification for possible cracks and hard spots in reinforcement welds were undertaken to avoid recurrence of cracking in this vessel and other similar in the plant. Thicknesses and support systems were set according to manufacturer specifications.
- Eliminating the reinforcement plate saddle in this case was non conducive; the next case studies, however, show how improper construction of these saddles and the difficulty to detect bad procedures once the vessel is built could severely reduce reliability.

## 3. Collector pipe in a natural gas turbine-compressor plant

Dehydrated natural gas at 63 bar flows through this discharge piping system at a natural gas turbine-compressor plant [6], see Fig. 5. A longitudinal fracture occurred in the weld joint between a 300 mm (12″) diameter collector and a 150 mm (6″) diameter venting pipe, see arrow in Fig. 5, after many years of service. Four hours before, discharge pressure of the compressors increased to 77 bar, for less than half an hour. This was due to a faulty valve. The section fractured in

**Fig. 5.** Case 2: Location of element failed in collector pipe.

two sections, both fractures initiated in the welded joint of the 6″ tube, a fillet welded flange with a circular reinforcement plate fillet welded to the collector.

Collector material has a SMYS of 400 MPa, probably an API 5L X56, SMYS for tube material only gets to 312 MPa. The upper end of the tube section adhered to the weld shows around 20% reduction of thickness due to plastic deformation. The root of the tube-to-reinforcement weld shows small tears and fractures that indicate lengthwise traction of the tube. Fig. 6a shows the failed section. The severe deformation after the failure, typical of pressurized gas containers following a blow up, makes it difficult to pinpoint the path for crack growth. The physical model in Fig. 6b allows defining fracture initiation.

As seen in Fig. 7, "fracture 1" is approximately longitudinal to the collector, and continues longitudinal in the tube. This "fracture 2" becomes circumferential and arrests when coming near to the bolted jacket of the tube, after crossing a distance of 150 mm. The other fracture in the collector is approximately circumferential. This "fracture 4" curves in the weld between reinforcement and tube, and continues circumferential in the tube. This "fracture 3" seens to cross fracture 2 and continues 20 mm, until arresting in tube material. Fracture 1 from point B until the reinforcement-to-collector weld is mode one (tension), ductile with shear lips. When crossing the weld between reinforcement and collector, the fracture is brittle, with chevron marks, see Fig. 8, and immediately becomes ductile again. Fracture 3 (sections B–E), took place in mode three (anti-plane shear), with signs of tear by loads normal to the surface of the plate.

Fig. 9 shows in detail zone B of initiation. Fracture initiation occurred at severe defects in the weld between reinforcement and pipe. Although somehow masked by corrosion, Fig. 9 shows a fish eye type of crack, named 'a' in the figure, which is typical of a hydrogen crack (cold crack during welding). Several pores coalesced to form another small crack, named 'b' in the figure. Fractographic analysis showed that these two cracks eventually coalesced to form a single surface crack, indicated by the dotted line. This first propagation was brittle, coalescence occurred either during the preoperational hydrostatic pressure test, or during the overpressure event.

To further elucidate this, a mechanical model of static stresses at the intersection due to pressure was carried out according to the procedures of section 9 of API 579 [7]. Initial crack depth $a$ is 5.18 mm, the depth of the lately evolved crack $b$ is 6.2 mm (crack "a" coalesced with pores). In a tube of fine wall internal pressure creates zero radial stress and a hoop stress according to the equation of Barlow. The presence of the 6″ hole generates a stress concentration. The reinforcement reduces the stress to levels similar to which would be without the hole, but previous pressurization of the reinforcement rises stresses again, so that a stress concentration of 2 is considered. For an inner diameter $D$ = 300 mm, a thickness of reinforcement $t$ = 7.2 mm, and an internal pressure of 6 MPa, the nominal circumferential stress is

$$\sigma_c = 2 \cdot P \cdot D/2t = 2 \times 120\ \text{MPa} = 240\ \text{MPa} \tag{1}$$

In absence of effective post weld heat treatments, maximum residual stress is around half of yield strength of base material (150 MPa) [8]. So maximum stress in the tube wall is

$$\sigma_c + \sigma_{res} = 240 + 150 = 390\ \text{MPa} \tag{2}$$

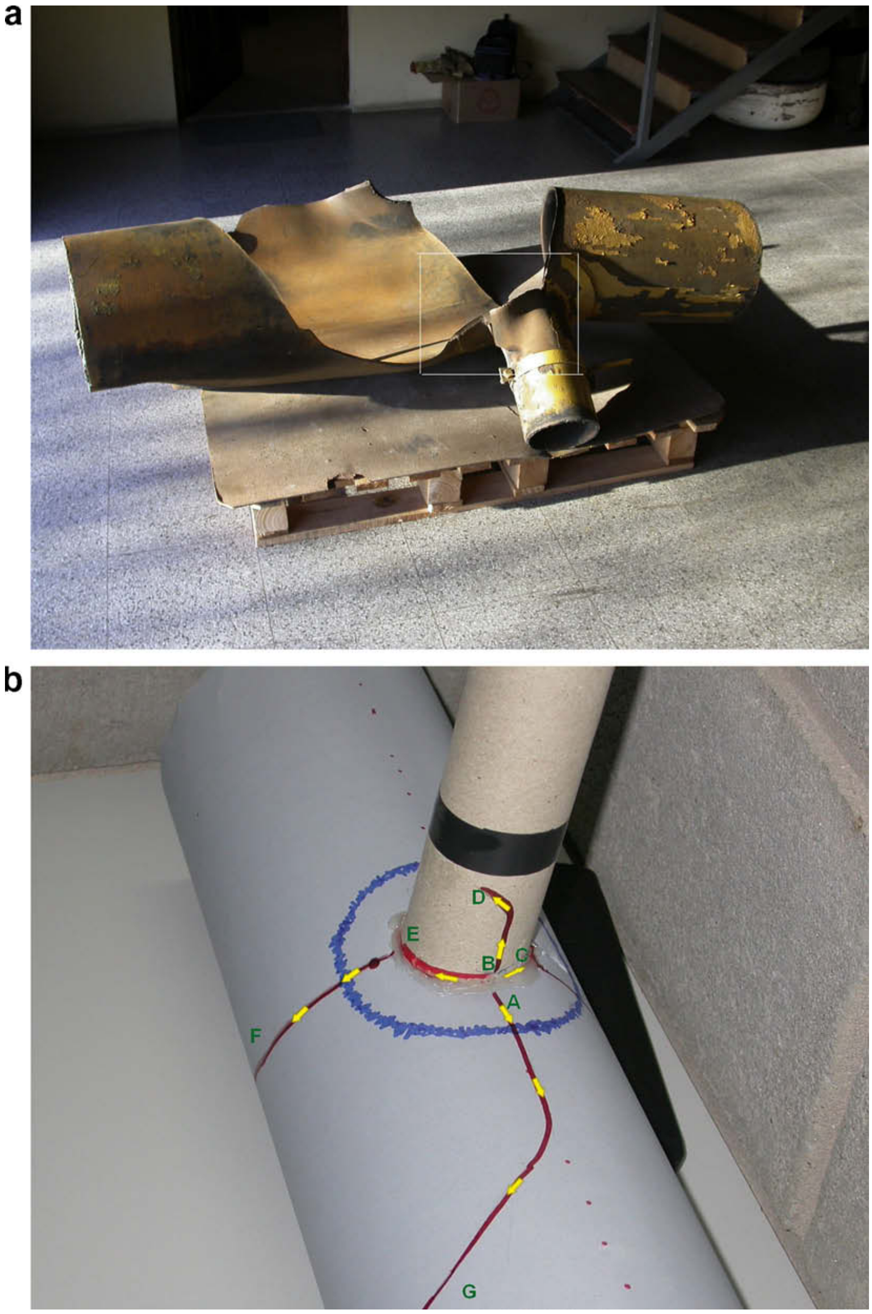


**Fig. 6.** (a) Section 12 of the collector, fractured into two sections. (b) Fracture initiation.

For a semicircular crack in the pipe wall with a depth of 80% thickness, the applied stress intensity factor *K is*

$$K = 1.2 \times 390 \times (3.1 \times 0.006)^{1/2} = K_{\text{capl}} = 73\ \text{MPa m}1^{1/2} \tag{3}$$

Taking into account the poor quality of manufacture, toughness of reinforcement material is likely low, conservative estimation is

$$K_{\text{IC}} = 100\ \text{MPa m}^{1/2} \tag{4}$$

Crack driving force to toughness ratio is

$$\text{Kr} = K_{\text{capl}}/K_{\text{IC}} = 0.73 \tag{5}$$

The applied stress to the section in the case of plastic collapse is mainly the Von Mises equivalent due to pressure. Using a factor for thickness reduction of section of 1.5:

$$\sigma_{\text{c}} = 1.5 \times 240\ \text{MPa} = \sigma_{\text{VM}} = 360\ \text{MPa} \tag{6}$$

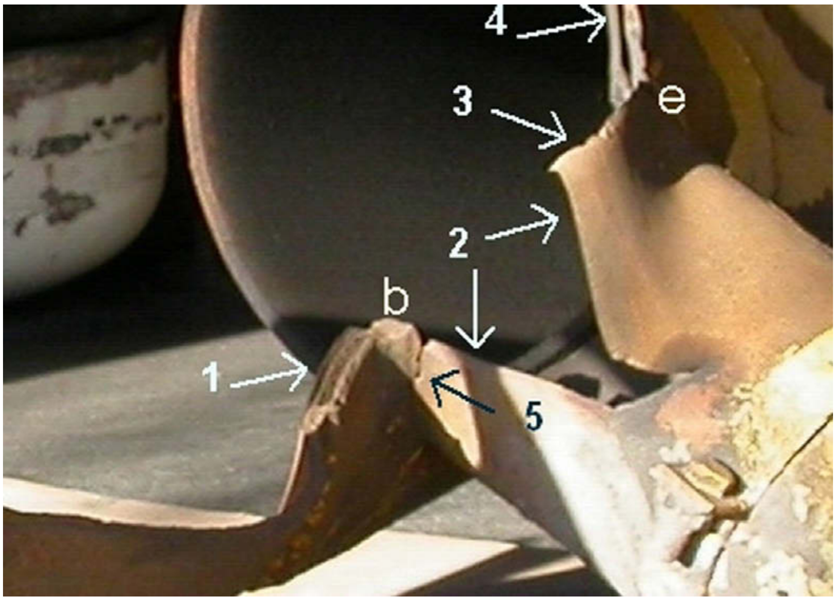


**Fig. 7.** Details of the zone box in Fig. 6a.

**Fig. 8.** Brittle fracture with chevron marks in the collector material.

So the applied stress to flow stress ratio is

$$Sr = \sigma_{VM} / \sigma_f = 0.76 \tag{7}$$

The point (Sr = 0.76, Kr = 0.7) falls in the safe part of the level 2 FAD of API 579, although near the safety limit.

The propagation of the failure from the preexisting weld defects can therefore be reasonably explained, only considering nominal internal pressure. Longitudinal stresses in the tube produced during assembly or during the previous overload could have generated larger stresses. Operation data allows to explain why the failure was 4 h delayed after the overpressure. It was found that the overload occurred in the late afternoon of a winter day, after dark, the low room temperature could have reduced material toughness, leading to crack instability and final failure.

Some of the root causes related to weld defects can be seen in this Fig. 9, but these were not the only construction flaws. The cross sections depicted in Fig. 10, for example, show numerous aspects of weld manufacture that do not match standard quality requirements: large pores, cracks, undercuts, weld toes of different size, etc. The welds between tube and collector and between tube and reinforcement would have to be rejected if evaluated under the requirements of, for example, API 1104, due to the presence of a great amount of large defects [9]:

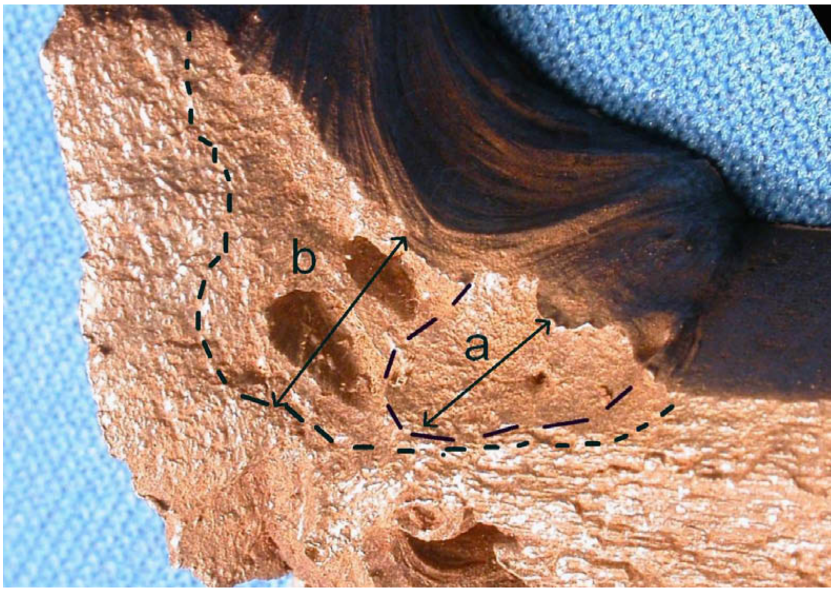


**Fig. 9.** Details of Zone B of initiation.

- The hole in the collector and the tube end were flame cut.
- The tube-to-collector and tube-to-reinforcement welds show great amount of pores and large undercuts.
- The end of the tube does not cross the hole in the collector pipe, but tops its outer surface; this is an error in the design of the joint.
- The fillet weld shows legs of different size, their sections in some cases are below the thickness of the welded elements.
- Arc strike marks were found throughout the surface of the reinforcement.
- The purge hole of the gap between collector and reinforcement was filled with a weld of very bad quality; the weld detached during failure and remained at one side of the fracture.
- The inadequate welding sequence caused craters at weld start and stop regions, and cold cracks.

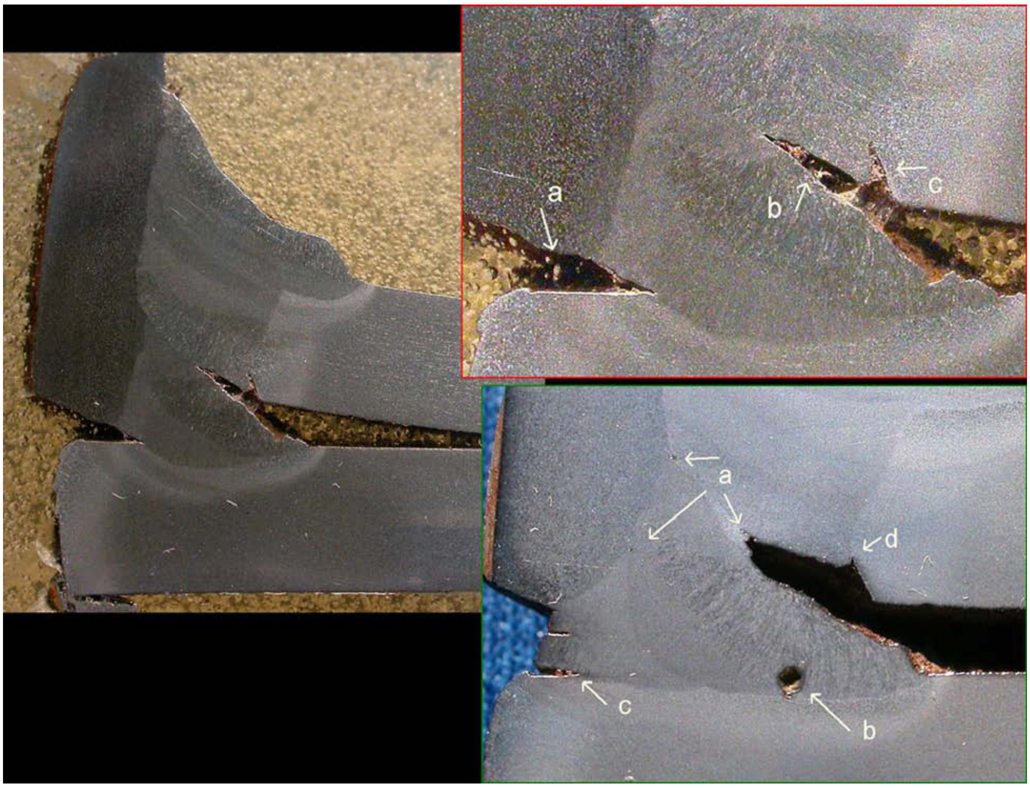


**Fig. 10.** Several cross sections of defective reinforcement.

Fracture mechanics analysis shows that these initial defects, however gross, were stable during subsequent service, for many years, until a small increase in pressure followed by cold weather re-activated propagation.

## 4. Suction vessels in natural gas alternating-compressor plants

Repeated cracking in plate reinforced welded flanges of several suction vessels in identical natural gas compressor plants presumed design, construction or operation problems [10]. In two of such failures investigated, cracks in the reinforcement saddle of the flange connections were detected before gas leak. Cracks initiate at the toe of the pipe to saddle weld, see Fig. 11, somehow similar to those found in Case 1 (Fig. 2). The root cause analysis of the problem included:

- Verification of design of the compressing facilities.
- As built drawings verified in field.
- Study of vibrations and pulsations [11].
- Failure analyses.
- Analysis of design assumptions of dampeners.
- Inspection procedure with indication of defect acceptance/rejection.

Evidences proved that, as expected, the cracking mechanism was mechanical fatigue, controlled by the cyclic variations of stress in the welded joints, with origin in variations of internal pressure and in dynamic phenomena related to vibrations and pulsations. The two analyzed failures have common characteristics, as depicted in Fig. 12: they initiated in the same zone in both large bottles; in both cases cracks started off large defects in the welds; the quality of the welded joints was found deficient.

In the outside, the reinforcement looked very well. Since according to ASME VIII Div. 1 these welded connections do not require NDT inspection, embedded defects stayed undetected. Cross sections (Fig. 13a) revealed many weld defects, in one case a weld bead over the entire surface of the reinforcement was made to comply with thickness specification (Fig. 13b).

Fatigue propagation of cracks require an initiator and a driving force, these are in this case some gross weld defects (hard spots, inclusions, and voids) and vibrations/pulsations of the system, respectively. A small crack in the outer surface of the container would have a very short fatigue life, further propagation would occur not only in the reinforcement but also in the vessel thickness, so immediate replacement of such flanges was required. Subsurface indications could be repaired by means of the replacement of the reinforcement or removal of indications and repair weld. The final solution to this recurrent problem was to repair the defective joints, using material and procedures according to standard best practices, including 100% inspection of installed large bottles, and to reduce the amplitude of vibrations in the bottles and associated pipes. This required some re-engineering of the system, as per ANSI-API Std. 618 [11].

In order to detect any other manufacture defects, particularly those preexisting defects from which cracks could initiate, non-destructive inspection of all potentially dangerous welds in all similar equipment was required. In all cases, critical

**Fig. 11.** Case 3: Crack initiation in the reinforcement saddle of the flange in suction vessel.

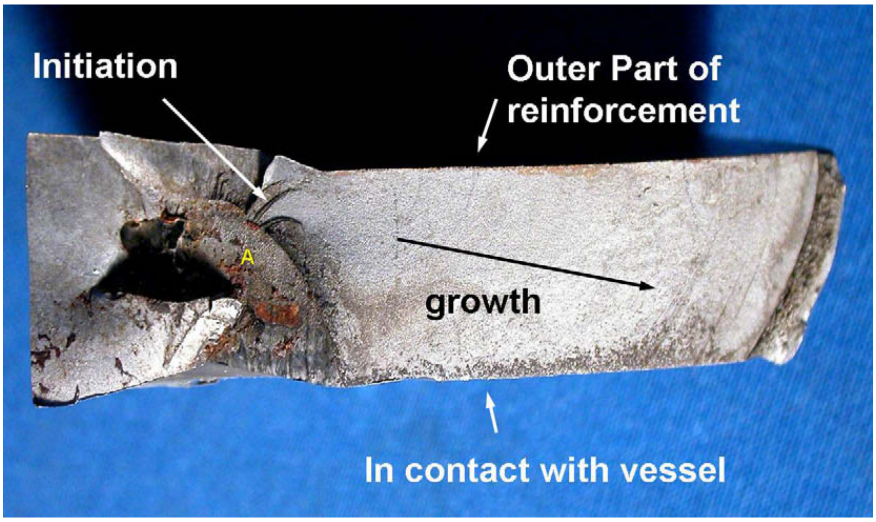


**Fig. 12.** Characteristics of the two analyzed failures in suction vessels.

regions for detailed inspections were pinpointed: those in the outer half of the reinforcement thickness, near the weld to the tube.

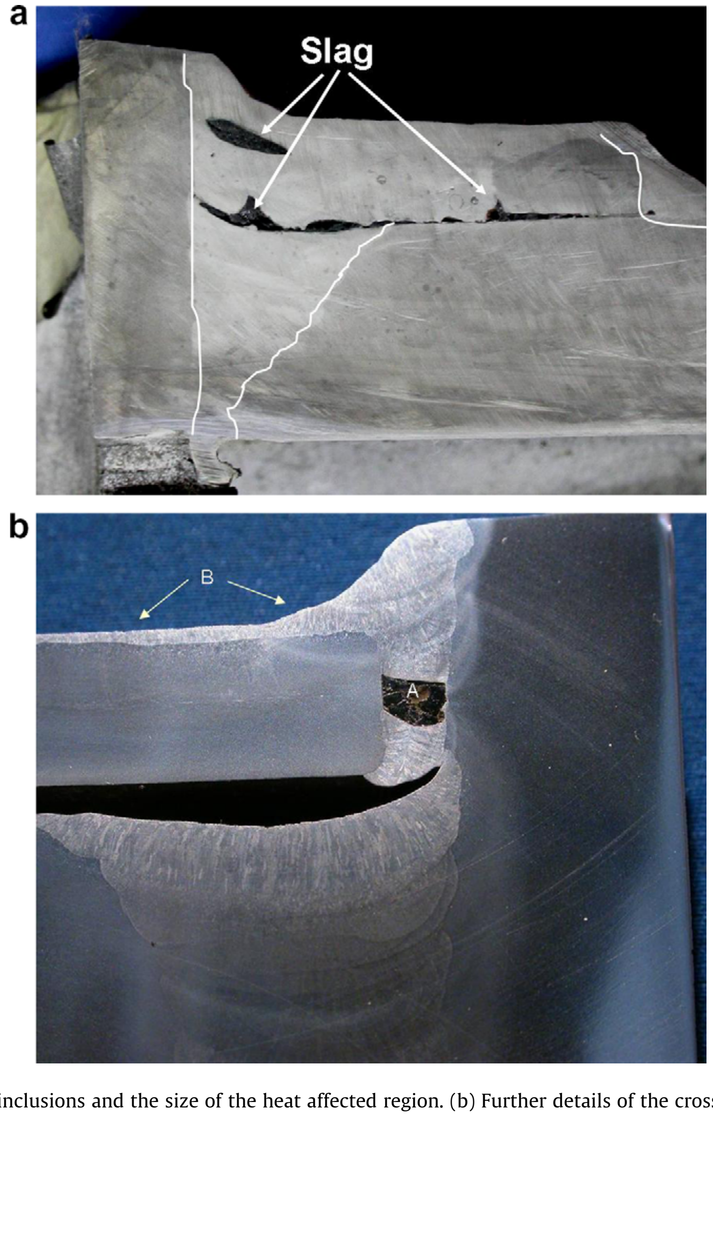


**Fig. 13.** (a) Cross section, note slag inclusions and the size of the heat affected region. (b) Further details of the cross section of the reinforcement welds.

## 5. Conclusions

Present rules in fabrication codes are aimed to reduce the probability of fatigue cracks in flanges welded to pressure vessels subjected to cyclic pressure or vibrations. Yet, several leaks and ruptures have recently occurred at flanges in pressure vessels and pipes. A review of some of these cases is presented, and their common root causes highlighted and discussed:

- Discharge vessels at an alternating-compressor in a petrochemical plant.
- 12″ collector pipe in a natural gas turbine-compressor plant.
- Suction vessels in two natural gas alternating-compressor plants.

NDT inspections, fractographic and microstructural analyses, and mechanical and chemical tests are briefly discussed, in the characterization of crack propagation into vessel material. Common root causes for the failures are investigated, and the influences of manufacture and operation conditions on crack initiation and propagation mechanisms are highlighted.

Possible mitigation measures are discussed in each case, and common conclusions yielded. Fatigue propagation is, as design expects, due to the large number of cyclic loads from gas pulsation and vibrations from the compressors. Some conclusions allow increasing mechanical reliability and fitness for service of these flanges, especially before entering service. The main aspect still is to assure an adequate dimensioning of the system to reduce the amplitude of cyclic stresses, but there are also some manufacturing and inspection aspects to address.

Common aspects of these failures are crack initiation in heat affected zone of welds, all related to an inadequate design of the reinforcement and of the welding procedure. Weld inadequacy is sometimes a result of large differences in thickness. Increase in thickness as an attempt to increase safety is found to be detrimental, since larger than expected cyclic stresses are sometimes due to excessive weight of the vessel. Improper foundation due to design errors or maintenance deficiencies leads to higher than expected vibration amplitudes. The use of fillet welded reinforcements induces rigidity inconsistencies within the flange joint, and concentrates deformations in the failed regions. In one case, it is believed that preoperational hydrotest could have been detrimental, by coalescing individual small weld defects into a crack.

Most cracks initiate from the outer surface, so an adequate non-destructive inspection program during service helps prevent failures. But other cracks initiate in the outer half of the thickness of the reinforcement, from very difficult to avoid weld defects. Changes in the design of the reinforcements is required. Ultrasonic testing, with an adequate procedure, can be reliably used to detect these defects.

Although some operation conditions have influenced the onset of crack growth, common root causes for these failures are related to poor manufacturing practices. Flanges with welded reinforcements are not a mechanically very efficient design. However, most relevant non-conformities are not related to inadequate design, but to bad workmanship and welding procedures. Gross weld defects were left in the welds between pipe and vessel. The reinforcement plate makes it impossible to detect defects in the pipe to vessel weld, during post construction and in-service NDT testing. In line with this, construction codes such as ASME PVBC section VIII Div. 1 do not specify a radiographic inspection.

Reinforcement plates are used due to reduced manufacturing costs, when compared to machined or forged transitions such as weldolets. Reasonably good results are most often obtained with this reinforcement design due to some fortunate factors:

- Good workmanship and inspection during fabrication.
- Hard to inspect weld areas are close to mid – thickness, that is, close to the neutral plane of the section when subjected to bending loads.
- Highest loads due to thermal transients and mechanical loads from the pipe and accessories fitted to it are of the bending type, so that the region most severely weakened by the welding defects is not subjected to high stresses.

Owner specifications should aim to avoid manufacturing vices, especially in critical regions where fatigue conditions due to vibrations and pulsations are foreseen, and where the consequences of a leak could be high. Inspection should play a key role during manufacturing of these joints, since NDT inspection after construction would not reliably detect gross weld defects.

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