Crack Analysis and Improvement Scheme of High Pressure Cylinder of CO_2 Compressor

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1. Introduction

The CO_2 compressor is one of the key equipment items in urea production. Its main function is to compress CO_2 from purification section in the ammonia plant and send it to the high pressure synthesis system in the urea plant. The compressor used in the new section of our Plant is of model 6MD32. Its main technical parameters are as follows:

Inlet pressure: 0.15 MPa	Outlet pressure: 15.7 MPa
Piston force: 32 t	Stroke: 360 mm
Gas flow: 187 m3/min.	Rotation speed: 300 r/min.
Number of rows: 6	Number of stages: 5
Shaft power: 3088 kW	Motor power: 3400 kW

The compressor has been put into operation after its installation in 2007. On March 5, 2010, overpressure of the fourth stage of the CO_2 compressor appeared. During overhaul inspection of the outlet valve of the fifth stage (aside the midbody), the inspector found penetrative crack in the valve cavity. A dye penetration test was carried out for confirmation.

2. Analysis on the cause of crack

2.1 Form of crack occurred

The crack is at the bottom of the cylinder valve cavity, with a 30° included angle to the axial line of the cylinder bore and the crack has penetrated through the valve seat. From the cracked position and adjacent machined end face, it can be seen that the forgings have obvious cavities formed due to the defect of internal loosing. Dangerous points including A, B, C and E of the valve cavity have sharp corners and cut marks without rounding processing. Moreover the roughness of the machined surface is relatively high, Rz25 (approx.). Refer to Figure 1 for details.

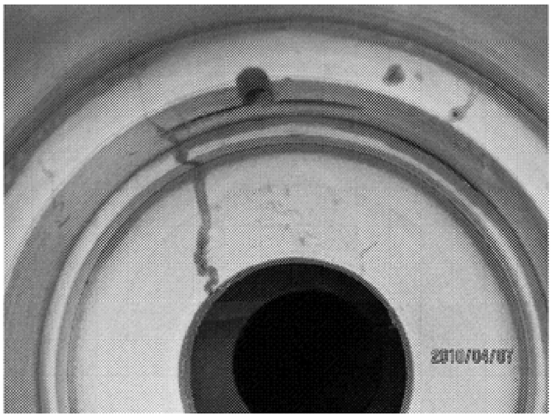

Fig.1 Crack in the valve cavity

2.2 Cause analysis

2.2.1 Load–bearing analysis

Outside of cylinder valve seat bears exhaust cavity pressure P2, while the inside bears internal gas pressure P1. Because P1 is variable during piston motion, P1 belongs to alternated load. Stress amplitude, mean stress and cycle do not change with the time during each circulation, so they are stable varying stresses.

The stress amplitude is: $(\sigma max-\sigma min)/2=38$ MPa

In addition, besides axial stress, radial stress and circumferential stress under alternated load, the bottom of valve cavity also bears preload of valve gland studs. Due to temperature stress caused by inlet/outlet temperature and cylinder cooling effect and other edge stresses, the load-bearing state is very much complicated.

2.2.2 Structural strength analysis

The valve is located at radial opening of the cylinder, while the cavity bottom is located at the crossing point of the two orthogonal cartridges (cylinder and valve cavity). Refer to Fig. 2 for specific structure of valve cavity. Therefore, affected by orthogonal circle deformation, valve cavity is deformation restraint at boundary. Moreover, as main dangerous points A, B, C and E (especially point B) bear force, the deformation is more complicated.

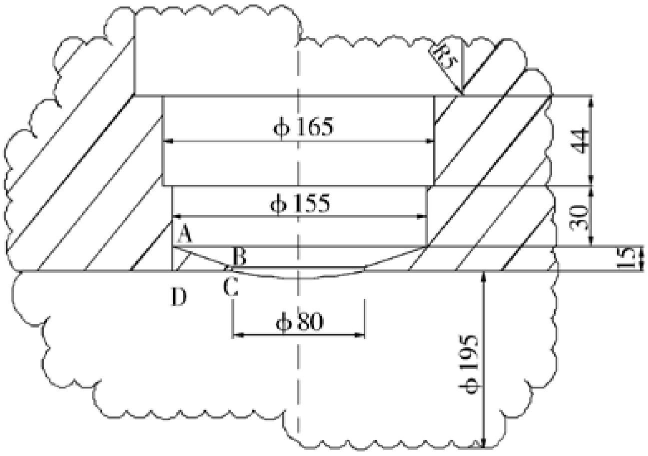

Fig.2 Structural diagram of valve cavity

In this article, it is believed that the thick-walled outer layer of the cylinder is affected by the partially thin-walled cylinder which is radically cut, and the influence due to deformation of valve cavity can be corrected by means of stress concentration factor. Stress analysis and calculation of point B are as follows:

Ratio between thicknesses of inside and outside diameters of valve cavity

$$K = \frac{D_i/2 + \delta_{min}}{D_i/2} = \frac{97.5 + 3.5}{97.5} = 1.04$$

Circumferential stress

$$б_{ti} = P_S \frac{K^2+1}{K^2-1} = 15.6 \frac{1.04^2+1}{1.04^2-1} = 405.6\,\text{MPa}$$

Equivalent stress

$$б_d = \sqrt{\frac{1}{2}(б_{ti} + P_g)^2} = \sqrt{\frac{1}{2}(405.6 + 15.6)^2} = 297.8\,\text{MPa}$$

Stress concentration factor

$$\frac{D_i}{d_i} = \frac{195}{80} = 2.44$$

Kt : taken as upper limit 2.5.
The stress corrected: σd0=Ktσd=2.5×297.8=744.58 MPa
Material of cylinder body is 35# forged steel. σs=467 MPa
Safety factor: ns=σs/σd0=467/744.58=0.63

According to the data provided in Materials and Structural Design of Displacement Compressor compiled by Professor Chen Yongjiang, safety requirements can be met only when n≥2, hence, safety strength requirements cannot be met based on what mentioned above.

2.2.3 Analysis on materials of forgings

(1) According to analysis and quality certificate, chemical compositions of the forgings comply with the requirements specified in Steel Forgings of Positive Displacement Compressors (JB-T 6908-2006).
(2) As to metallographic structure, the matrix is ferrite and pearlite, and grain size is in the range of grade 4-5.
The Widmannstatten structure in most view fields is ≤ grade 1, while a few fields have the structure in the range of grade 2-3. Ferrite is needle-like distributed in grain boundary and prone to expansion into grains.
In the matrix, a few view fields have Widmannstatten structure in the range up to grade 2-3, exceeding the range of grade ≤1 specified in Determination of Microstructure of Steels (GB/T13299-1991). Such structural defect may impair bonding force between grain boundaries, causing microcrack under alternated load, thus further resulting in metallic fatigue.

The hardness of the said materials tested is in the range of 108-150HB, not complying with requirements specified in Steel Forgings of Positive Displacement Compressors (JB-T 6908-2006). Average hardness is lower than standard requirements, which badly affects strength index. Moreover, the higher hardness difference reflects greater structural inhomogeneity.

In conclusion, the main reason of such crack is that the structural strength safety factor is lower than the allowable safety factor. In addition, defects of partial metallographic structure, structural design and machining cause stress concentration. Besides, the machining accuracy is lower; the defective machining parts appear microcrack and further spread into penetrative crack under long-term action of alternated load.

Affected by structure and surface machining accuracy, transition section between cavity bore and bottom bore becomes a stress concentration area. In the loose defective area of forgings, metallic continuity is spoiled. In addition, the defect of metallographic structure has impaired the bonding force between grain boundaries. Under the action of alternated load, materials at locations where stress and defect are relatively concentrated undergo shearing slippage, thus forming a cracking source. As fatigue crack spreads gradually under the action of alternated load, effective section of components reduces while stress increases gradually. The equipment is structurally designed with lower safety factor and lower strength, which accelerate spreading speed of crack and finally the crack penetrates through the surface of the valve seat, resulting in short circuit of gas passage and overpressure of the fourth stage.

3. Improved scheme for purchase of new cylinder body

(1) In the technical agreement on procurement, put forward technical requirements on forgings of cylinder body and machining quality according to Steel Forgings of Positive Displacement Pressure Vessels (JB/T 8908-2006) and Technical Specifications of Large-sized Reciprocating Compressors (JB/T 9105-1999) and add the review and conformation required for reinspection of forgings provided by Party B.
(2) Thicken bottom of the valve cavity. With reference to the thickness value of HP cylinder valve cavity of Germany Boerger Company, define the value δmax to be 50mm and carry out strength reinspection. Define the safety factor to be 4.2m and reserve sufficient safety allowance. Meanwhile, adjust the sizes of related valve sleeves accordingly.
(3) Enlarge radius of R arc at each dangerous point. According to the results of test conducted by J.C. Gerdeen and R..E. Smith, stress concentration factor is minimized when $r/Di=0.35$, based on which corresponding R value can be calculated so as to alleviate stress concentration.
(4) Improve machining process of the cylinder. Add annealing treatment for stress relieving after rough machining to relieve machining stress in time.
(5) Improve machining accuracy of each machining surface of valve cavity. Fine boring is required to be used at last. Machining roughness of each dangerous point is defined to be Rz1.6, while that of the spigot of the valve seat is Rz3.2.
(6) Apply special rolling and compressing treatment to each machining surface inside of the valve cavity so as to improve fatigue strength accordingly.

4. Conclusions

The valve cavity of the HP cylinder is most prone to failure. According to data related, it is known that cylinder cracking caused by unreasonable design accounts for 74% of total cracking accidents. Therefore, the design department should fully recognize the importance of such problem, optimize structurally to avoid stress concentration and reserve sufficient safety factor allowance. Finite element analysis and calculation should be carried out if possible. Manufacturers should carry out warehousing quality inspection strictly to guarantee the quality of forgings, conduct machining as per requirements of drawings to avoid stress concentration of key components during the machining process. Only by giving common concern from all parties concerned and being strict in product quality, can the reoccurrence of such accident be avoided.

Translator notes:
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