Studies on some Aspects of Conveyor Drive Pulley Design

Mr. Ronak R Patel*
* - Lecturer, Mechanical Engineering Dept,
Birla Vishwakarma Mahavidyalaya
VV Nagar, Gujarat, India

Prof. Shashank P Joshi¹, Prof. P. M. Agrawal²
1,2 - Associate Professor, Mechanical Engineering Dept,
Birla Vishwakarma Mahavidyalaya
VV Nagar, Gujarat, India

Abstract—The present paper brings out the inadequacies of the standards and codes used for conveyor drive pulley design. Some of the aspects of conveyor drive pulley design not covered in standards and codes have been evaluated in the light of the shaft, type of diaphragm and shell with an integrated approach. The different design approaches like classical mechanics approach, finite element method and modified transfer matrix have been discussed highlighting their pros and cons.

Keywords— conveyor drive pulley design, integrated approach, classical mechanics approach, finite element method and modified transfer matrix.

I. CHALLENGES IN CONVEYOR PULLEY DESIGN

Present day conveyors use drum type welded steel construction, comprising of a shaft, hubs, diaphragms (also termed as discs or webs) and shell or rim (to support the belt width). The selection of conveyor drive pulley is based on tension range of belt transmission [1]. Usually, for the lower tension range up to 1300 N/cm of belt width, “conventional or standard construction” pulleys (with welded hub fig.1) are used. For the tension range of 1300 to 4500 N/cm of belt width, “Engineered Class” pulleys (with welded fig.1 or integral hub fig.2) are used and for the tension range of 4500 to 17,500+ N/cm of belt width, “T-section” pulleys (with integral/ profiled hub fig. 2) are used.

Figure 1. Conventional/Standard construction pulley

Most of the general conveying applications, fall within the range of lower tension ratings. The welded hub and diaphragm type pulley design (conventional) is chosen for lower tension ratings, due to the relative ease of manufacture. However, for higher tension ratings, the economic design is achieved through other types of constructions. Conveyor pulley has been recognized as a critical component of conveyor, failure of which, can lead to extensive damage and down time and hence conveyor pulley must be designed carefully for the successful operation of conveyors.

The conventional conveyor drive pulley design is based on conveyor belt tensions of tight & slack side and self-weight under steady state conditions. Different component parts of the drive pulley like shell, diaphragm, hub and shaft are subjected to bending stresses. Thus, strength and rigidity are the main criterion of design. General practice in design is to use classical mechanics approach for strength and rigidity and to use higher factor of safety (~3 to 4). Design is checked for permissible deflection (rigidity) and bending stresses.

A conveyor pulley is a rotating device under dynamic, asymmetric lateral load, for every revolution all its parts go through a complete reversal of stresses caused due to load. This leads to fatigue in the components. Hence, when designing a pulley, the problem is to provide sufficient strength in the shell, diaphragms, hubs and shaft to prevent failures due to high stresses, and, at the same time, provide just sufficient flexibility in the overall structure so that it neither fails due to fatigue, nor wobbles on the shaft. It

Figure 2. Turbo diaphragm/T-section/taper profile diaphragm
appears that to arrive at the optimum design, all the parts comprising of the integral unit of the pulley should necessarily be designed together.

Codes of Practice and Standards used for designing conveyor pulley are DIN, IS, ISO, BS and CEMA (Conveyor Equipment Manufacturing Association). However, it is a common practice for industries to adopt their in-house design and manufacturing procedure following the standards, but guided by their own experience. Unfortunately, most of the codes on pulley design do not adequately treat the interdependence of stresses and deflection [8]. While the CEMA specification on pulley design takes account of fatigue life of shaft, it provides no guidelines for obtaining the thickness of shell and diaphragm. The MHEA(Material Handling Equipment Association) practice allows thin web construction but its expression for evaluating shell thickness appears rather conservative as it ignores the influence of wrap angle on rim stresses. Although it links up the shaft deflection with diaphragm stresses, it does not interrelate the stresses in shell and diaphragm and their connectivity. Moreover, it does not deal with the tapered diaphragm construction. The other codes like IS: 8531-1986, BS 2890 and DIN 22101 provides overall dimensions for pulleys, without specifying any technique for sizing the individual parts.

This is because of the limitation of classical mechanics approach that deals with design of elements of pulley separately, without considering the effect of inter-connections (behavioral change) and inter-related stresses arising from interconnection. Hence, it is necessary to analyze the stresses in different parts of a pulley and their inter-relationship. This requirement opens an avenue for improving design procedure with another approach like finite element analysis (FEA).

Finite Element Analysis can analyze the stresses in different parts of pulley and their inter-relationship, which in turn can bring about improvement in design and saving in material cost. While, pulley design can be carried out as per selected standard, the analysis tool helps to optimize the design and can be beneficial in assuring reliability.

Based on pioneering work carried out by Lange [3], Schmoltzi [9], Das, Pal-Hong-Sheng [2] and Longman [5] have investigated pulleys using analytical means. Most of the analytical researchers considered pulley part-by-part rather than as integral unit. Although some finite element studies have analyzed the pulley as a unit, not much is known about their loading condition, assumptions and boundary conditions [8]. Moreover, no systematic study on the influence of changes in one part on another has been noticed. It appears that an analytical approach can be very useful to supplement the present state of knowledge but for reliable design of pulley, finite element study will be beneficial.

The web thickness varying along its radius needs to be optimally calculated. There is no well established formulation for such designs. Location and thickness of web is according to the designers discretion. FEM is used for an integral analysis of pulley, useful in suggesting certain basic guidelines. It supplements the current knowledge available in various standards and code of practice in the field. FEA tool helps to optimize the design and improve reliability.

II. DESIGN CONSIDERATIONS FOR CONVEYOR DRIVE PULLEY

Designers of conveyor pulley must design the pulley to perform the required function with high strength to low weight ratio for its rated period of life. At the same time, it must be “design for manufacture”. The optimum solution to a design requirement will be accomplished, if the designer takes into account the following considerations.

1. Total service requirements: Tension rating, Service life, Working temperature and Wear & tear.
2. Type of loading:
   a. Conveyor pulley is subjected to radial pressure along the circumference of pulley shell due to tensile forces in the belt [7].
   b. Bending load on diaphragm due to belt wandering.
   c. Centrifugal force (predominant at higher belt speed).
   d. Impact due to starting, stopping and braking, impact is more significant for higher lump size and bulk density.
   e. Wear on shell (predominant for high tension pulleys).
3. Type of stresses developed in various component parts.
4. Mechanical & physical properties of material being used for Conventional Pulleys.
5. Construction of pulley and fabrication techniques.
6. Quality Specification and Inspection Techniques:

   Appropriate standards & codes shall be used for qualification of pulley design and manufacturing. However, manufacturers are using their own quality specifications developed for their applications. Here, it is worth noting that there is a lack of universal design standard (table 1)[13]. Because of the diversified requirements, the three main design standards vary significantly. It is difficult to set standard quality specifications for every typical application.

<table>
<thead>
<tr>
<th>ISO 5048</th>
<th>GOODYEAR</th>
<th>CEMA</th>
</tr>
</thead>
<tbody>
<tr>
<td>T1 = 934 KN</td>
<td>T1 = 945 KN</td>
<td>T1 = 912 KN</td>
</tr>
<tr>
<td>T2 = 311 KN</td>
<td>T2 = 315 KN</td>
<td>T2 = 309 KN</td>
</tr>
<tr>
<td>Te = 623 KN</td>
<td>Te = 630 KN</td>
<td>Te = 608 KN</td>
</tr>
<tr>
<td>P = 2959 KW</td>
<td>P = 2995 KW</td>
<td>P = 2888 KW</td>
</tr>
</tbody>
</table>

Table: 1: Conveyor type is inclined, with length=1532 m, capacity = 2281 TPH and belt speed 0f 4.75 m/s and table content compares the different industrial standard calculations.

The following articles discuss some of the aspect of important design criterion and approaches used to design the conveyor pulley.

III. STATIC STRENGTH AND RIGIDITY CRITERION

While evaluating ductile materials, yield strength of the material is usually used as the failure criterion. Sethi & Nordell [10] has applied static energy theory for performing static strength analysis. The Von Mises stress criterion is used in the theory and he has assumed a biaxial stress state for
simplicity and used a multiplier of 0.7 which accounts for uncertain condition i.e. porosity, inclusion etc. This multiplier of 0.7 is slightly higher than 0.6 or 0.66 multiplier used for welded structure. The range for maximum acceptable Von Mises stress in shaft, end disk and shell is 0.7 * yield stress[10]. For the shaft sizing, value of principle stresses can be found from BS 153 parts IIIb & IV:1972 or one can use BS 970:1972(EN3A).

As pointed by Ravikumar & Chattopadhyay[8], the stresses & deflections of pulley components and their interconnections being inter-related, the design can be handled in several ways.

a) The shaft can be made so rigid that even without any stiffening effect from the webs; it will not deflect too much to cause severe stress reversals.

b) The pulley can be built with more than two hubs so as to stiffen the shaft and prevent any excessive bowing.

c) The pulley can be made so heavy and rugged that it will not yield to bowing of the shaft even with two hubs.

d) The pulley can have the webs so flexible that they are not stressed beyond the limit in spite of whatever flexing takes place in shaft.

Of these methods, (a) is used most commonly today. The use of method (b) is almost obsolete, being confined to cast iron pulleys. Extra heavy-duty pulleys, following method (c), are available, but not popular. The last, (d), is the most economical but the design of the pulley is more complicated and needs to have in-depth knowledge of the stress distribution and deflection in all the parts and their interdependence.

Ravikumar & Chattopadhyay [8] has adopted the method (d) & developed software for the analysis & design of pulley based on FEM. Nevertheless were not able to ensure near permissible stresses throughout the profile of tapered web and thereby reducing the pulley weight and at the same time increasing fatigue life of the profile web. For the wider belts the keyed shaft proves more economical & shaft diameter influence the stresses in other components of pulley [8]. Also he has worked on the effect of providing stiffening rings inside the rim and stresses in tapered web profiles.

IV. FATIGUE STRENGTH CRITERION

In case of most pulleys, the largest range stresses in the shell are usually in tangential or hoop direction and occur close to the centerline of the pulley. Pulley with wide shell faces may have the largest range stress in axial direction due to bending in a region close to the shell-disk connection.

BS: 5400 part 10 is used to determine the allowable stress range for circumferential and seam weld in the shell for infinite fatigue life. The largest fluctuating stresses in the disk[5] are in radial direction and are due to end disk bending. The fatigue strength criterion used is that the maximum stress should not exceed the endurance stress, for infinite life. Vinits[5] used conservative endurance stress of 40% of yield strength(20% of shear) because of dynamic loads, unlimited starts and stops and irregularities in lagging thickness. For shaft, Fatigue range can be found by using modified Goodman diagram.

V. MANUFACTURING CRITERION FOR ELEMENTS OF PULLEY

The industrial standards as set forth by CEMA,DIN,BIS,ISO and belt manufacturer’s handbook gives a broader view about manufacturing criteria for various components of pulley excluding taper web profile.

A. Shaft:

The shaft should have an overall surface finish exceeding 250 RMS. All surfaces should be finished "bright," removing all scale and surface flaws. All fillet radii should be finished to 63 RMS [6]. Many design criteria refer to the shaft deflection limitations. The deflection refers to the angle made by the loaded shaft off its neutral unloaded centerline at the pulley end disk centerline. It is normally expressed in a dimensionless slope or tangent of the angle. All shaft steps or turndown sections from the pulley hub diameter to bearing journal diameter have fillet radii in the step transition sized to control the stress raisers.

B. Hub:

The hub being of massive construction has high rigidity. Its sizing is made according to the type of fasteners employed between shaft and hub. In the key fitted hub, it is difficult to dismantle hub having key fitted with shaft. In addition, hub may prove cheaper as compared to locking assembly.

C. Diaphragm:

End-disk connection at the hub and shell should have generous fillet radii. The general metal finish outside the fillets is specified to equal or exceed 125 RMS [10]. The end-disk weld termination, shown in fig.3 (a), requires full length welding on the inside and outside of the end-disk; welding only from the outside produces a significant stress raiser at the root of the weld. Numbers of shell failures were observed due to this condition [12].

The hub boring operation should be done after welding the web to the hub and stress relieving. The shaft should then be mounted, if possible, and the final machining completed on the shell. The balancing weights should not be welded to the web. They may produce unacceptable stress raisers and reduced web load bearing capacity.

Figure 3. Most common types of diaphragm construction

Fig. 3 (a) shows a pulley with a plate type end-disk that is welded to both the shell and the hub. Large alternating bending stresses combined with possible weld inclusions, high stress raisers in welded surfaces, restricts its application for high-tension ratings. Weld zones in pulleys are inherently more failure prone. Therefore, for engineered class pulleys, fig. 3 (b) shows that integrating the hub and turbine shaped
end-disk from a common metal plate minimizes welding. Fig. 3 (c) shows the shell outer section is incorporated into the hub-end-disk from a single metal form in more advanced designs. This eliminates all fillet welds. Fillet welds have a significantly lower allowable fatigue stress range than does native metal, typically 33% [British Standard BS5400 Part 10, 1980] of the metal's fatigue rating. The end-disk hub to disk fillet welds are subjected to usually compressive alternating stresses due to the locking device expansion.

D. Shell:

The belt is wrapped around the shell surface. Machining the shell and boring the hub after all welding and stress relieving are completed controls run-out tolerances and metal distortions.

Sethi & Nordell [10] has also given manufacturing criteria for various components of pulley. The shell outside diameter should be machined with respect to its bearing journal with 0.5 mm TIR (total indicator reading) for drive pulley. This will help to minimize excessive local stresses in steel cord belt splice due to eccentricity and load changes of the motor from variation in belt speed. Maximum acceptable belt misalignment is about 6% of its width, which can increase 10% bearing reaction on one side. He has also considered this in shell criteria. Further shell should have a uniform plate thickness after machining. Not more than 10% difference between minimum and maximum thickness is allowed. (Assuming minimum thickness meets stress criteria).

Shell circumferential weld have an allowable stress range of 77 Mpa (class C weld) and allowable hoop stress range of 100 Mpa (class B weld). This values apply, if the welds are full penetration and free from defects. If they are not ground flush and proven free of defect, allowable stress reduces to 55 Mpa (class D weld) and hoop stress range to 77 Mpa (class C weld). Shell axial seam weld has allowable stress range of 100 Mpa and hoop stress range of 77 Mpa (full penetration and defect free). If not, allowable axial stress reduces to 77 Mpa and hoop stress to 55 Mpa. These allowable stress ranges are for 10 million load cycles with 07% confidence level. Radiographic and/or a full ultrasonic inspection must be performed to evaluate the welds.

VI. ANALYTICAL TOOL THAT RESOLVE STRESS

Costly failure in large conveyor pulley had led designer to seek detailed stress analysis. To date, three types of approaches for pulley stress analysis have been reported in literature.

- Classical Mechanics Approach
  Developed by “Lange & Schmoltzi” [3, 9]
- Finite Element Method
  Developed by “Vodstrcil, Daniel & Sethi” [14, 10]
- Modified Transfer Matrix
  Developed by “Xiangjun Qiu & Vinit Sethi” [7]

A. Classical Mechanics approach

It is an approximate analytical approach, providing a closed form solution for stresses in a pulley. Lange [8] has done pioneer work in conveyor pulley design using classical mechanics approach. His major work was to find some basis for calculation and design of conveyor pulley elements particularly shell and diaphragm.

Flexible diaphragm absorbs small bending moment and almost all the moment is supported by shaft. If the diaphragms are rigid, then almost all bending moment is supported by diaphragm and it could be possible to remove central part of the shaft.

The maximum bending moment acting on shaft is at diaphragm for both types of pulley i.e. Drive and Non-drive pulley. For small load the admissible deflection method must be employed (Non-drive pulley) and for larger load the admissible stress method must be employed (generally for Drive pulley).

The most complicated part of a pulley and the area most prone to failure is the web or end disc or diaphragm [2]. Stress is not the only consideration. Shaft must be designed to handle deflection as well as stress. Excessive deflection will lead to problem not in shaft but in the part that the shaft supports. The problem is manifested through the connection to the shaft. In a pulley, the deflection of the shaft causes bending moment being applied to the shaft connection and the diaphragm.

Published investigation for some profile of tapered web being few, a demonstrative study needs to be carried out. An optimization study should be undertaken to ensure near permissible (uniform stress) throughout the diaphragm, thereby reducing the pulley weight and at the same time increasing fatigue life of the web. Steeply inclined or long conveyors have high reactive pull. Such conveyors are equipped with fabric belt of high-tension rating or steel cord belt. These belts impose severe load on pulleys. The conventional pulleys of foregoing description have certain weak areas, because construction is not derived on the basis of ideal design, but has come as convention due to other reasons. As against this, the turbo-diaphragm type pulleys have evolved as a better design and to avoid failure. The pulleys with turbo-diaphragm are used for highly stressed application and therefore, the choice of materials and qualities of workmanship are of very high standard. A characteristic of this type of diaphragm is in the welding of the diaphragm to the shell, where the alternating fatigue stresses are far lower. One of the load acting on shell of conveyor pulley is axial stress produced by lateral displacement of belt, this stress is small in comparison with other load acting on pulley and so it is neglected in Classical approach. The disadvantage of this method includes that Stress solution is not accurate at location near connection region between shell and diaphragm because of its poor approximation in treating elastic coupling between these components. Specifically the displacement of diaphragm and shell are not coupled at their connection. These lead to significant errors in stress held about connection. Also Stress solution at Hub and diaphragm connection is also not accurate. In addition, each and every force acting on shell is not considered.
B. Finite element method
   It is an accurate stress solution method. It has just opposite feature of the Lange Classical method. By the help of FEM, it is possible to design variable thickness diaphragm or tapered profile diaphragm but it needs experimentation. Advantages of the method include ease of treating complex geometry and boundary condition, accurate stress analysis and considerable saving of materials. However, computation is time consuming, selection of type of element and shape function requires trial and error or past experience and accuracy of the results mainly depends upon the generation of finite element mesh and boundary conditions imposed. Literature survey revealed that FEM has been used mainly for high tension rating pulley [11]. FEM has been used by Vodstrcil[14] for high tension rating pulley and Siva Prasad and Radha Sarma[11] for pulley shell.

C. Modified transfer matrix
   This method is based on a reformulation of transfer matrices for the pulleys cylindrical shell, diaphragm and shaft by using finite element concepts. It combines the strength of both classical stress analysis methods and finite element methods. Although there is limitation of transfer matrix method in handling complicated boundary condition, such as the boundary conditions for a pulley, but the solution obtained by Qiu and Sethi [7] have overcome this limitation by reformulating transfer matrix using finite element concept.

   The reformulated transfer matrix is essentially a special finite element, called transfer matrix based (TMB) finite element. The new method using these special finite elements in a model is known as Modified Transfer Matrix. It is capable of solving a class of structural elasticity problem (including pulley), whose governing differential equations can be reduced to a set of ordinary differential equations. Modified transfer matrix method is capable in handling complicated boundary condition, such as the boundary conditions for a pulley and the solution obtained by this method is generally very accurate.

   Accuracy of the method depends upon the number of transfer matrix based (TMB) finite elements (accuracy increases with higher number of TMB finite element) used in the model. Higher number of TMB finite elements in the model makes computation time consuming.

VII. OUTER DIAMETER TO SHAFT DIAMETER RATIO
   Leo J Laughlin[4] has developed a balanced design philosophy and evaluated with FEA. According to him [4], in past belt was made from cotton fabric carcass. So pulley diameter were quite large compare to shaft diameter, which gave pulley end disk to flex (he adopted method (d) according to Ravikumar), thus allowing hub to follow deflection of the shaft. The ratio OD/SD, the pulley diameter to shaft diameter, is a measure of the flexible nature of end disk. In past, OD/SD was approximately 6 to 8. Modern days belting is more flexible and stronger due to high strength synthetic carcass and allows the belt to conform to a small pulley diameter, at the same time the shaft diameter is being increased, due to higher tensions, So OD/SD ratio becomes smaller. As pulley diameter is small, cost of the drive pulley and packaging reduces and thus overall cost reduces even though shaft diameter is large.

   Due to the advancement in belt materials modern days OD/SD is in the range of 3 to 5. This tends to be a limiting value for welded steel pulley. As this ratio decreases pulley tends to be rigid and so is the end disk. This change in the geometry puts more pressure on the shaft connection and end disk and may lead to premature failure. Laughlin has found that OD/SD ratio should lie within 4 to 8 range based on the stress plots in components of pulley. This is very simplest way for a designer to ensure the flexibility in end disk in the design of pulley, which is often overlooked.

CONCLUSION
   Even though many standards and publications are there for conveyor pulley design but they are giving guidance for designing individual parts of pulley. It is necessary to analyze the stress in various pulley parts and their effect on other components part of the pulley and their interdependence of stress and inter-relationship to various parts of pulley.

REFERENCES