Cardan shaft design pdf

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Our technician will oversee the commissioning of new plants or single machines, providing the expertise and equipment needed to safely launch. Costruzione Allunghe e Trasmissioni s.r.l. Designing and manufacturing gearboxes, scoops and gimbal shafts for transmission. Universal ramparts. Transmission connections. Plates of lamellar scoops. Special joints and bearings of white metal. Transmissions for industrial use. Geary and pinions. Special joints. More GEWES Cardan Shafts, or even called u-joints, facilitate the reliable transmission of torque between spatially remote drive and exit trains. GEWES shafts offer suitable mechanical drive solutions in almost all industrial sectors because of their versatile design and their high efficiency. Our weight-optimized, energy efficient, high-performance universal shared shafts have been developed Using advanced FEM techniques and calculations to provide optimal tube wall strength and diameters for high xero and resistance bending. We use hardened steel and hardened steel for our u-joints. Calibrated high-precision steel tubes are used for particularly complex solutions. All materials used meet the requirements for maritime classification and the use of railway vehicles. This ensures the reliable and continuous operation of our high-powered cooling joints. Our Cardan shafts are characterized by low component maintenance and low maintenance costs throughout the product lifecycle. Our dynamic balancing systems allow us to balance universal collaborative shafts with high accuracy in accordance with the specifications for applications that include rapidly changing universal connections. Running with low vibrations reduces the load on the drive and universal joints. This results in a longer lifespan and creates less operating noise. Our dedicated staff are happy to help you find the right one-size-fits-all joint for your app or develop a suitable solution to meet your specific requirements. Here's a review of our series of Cardan shafts. Please click on the relevant line in the following table for details. SeriesLimiting torqueAlternating torqueSwing diameterFlange connectionDINFlange connectionSAEFlange connectionKV/XSFlange connection Face key[Nm][Nm][mm][mm][mm][mm]153501006058/65---3011003209075/90/1001120/1300--43240010009890/100/1201120/1300/1400100-5342001300115100/120/1501400/1500120-6362001700125120/150/165/1801500/1600120-5888002500155150/165/1801600/1700/1800152-68115004000160150/165/1801600/1700/1800152/180-70170005100174180/2251800--72210005100170180/2251800180-73250007300178180/220/225/2501800/1880180-772800011000204180/225/2501880180-7934000-204--200-803300013000215225/250/2851880/1900--834000018000250250/2851880--845500023000265285---855500023000250250/285/315---865800024000250285/315---9012000045000285285/315/350---9517500058000315315/350/390---9720000070000350350/390/435---9820000070000370350/390/435---82-H5500023000225225/250--225/25086-H10500036000250285/315--250/28590-H15000053000285315/350--285/31595-H21500075000315350/390--315/35097-H260000100000350390/435--350/390S1-H370000140000390435/480--390/435S2-H600000225000435---435/480Flange with Hirth Serration Flange with Cross Tooth Serration... can be found here. Please ask if your connections are not listed. We can offer you various special flank connectionscardan shafts of supersort gimbal shaft flank joints companion flanks of universal joint cross assembly 1. PROJECT REPORT ON DESIGN ANALYSIS OF UNIVERSAL JOINT FOR ROLLING MILLS is presented by THE TUKADOJI MAHARAJ NAGPUR UNIVERSITY NAGPU (M.S.) in a partial bachelor's degree in CODIURSION, presented by ANKIT ANAND (22) M. M. (39) MOHIT KATHOKE (46) SUGHOSH DESHMUKH (68) WASUKARNA JAIN (74) LED BY PROFESSOR G. R. NIHADE 2013-2014 DEPARTMENT OF MECHANICAL ENGINEERING SHRIDEOBABA COLLEGE OF ENGINEERING - MGMT. NAGPUR - 440 013 2. PROJECT REPORT ON DESIGN ANALYSIS OF UNIVERSAL JOINT SHAFT FOR ROLLING MILLS is presented by RASHTRACANT TUKADOJI MAHARAJ NAGPUR UNIVERSITY NAGPU (M.S.) in part-time bachelor's degree in CODIURIC, REPRESENTED BY ANKIT ANAND (22) M. MUKESH (39) MOHIT KATOTE (46) SGOSH DESHMUH (68) WASUKARNA JAIN (74) LED BY PROFESSOR G. R. NIHADE 2013-2014 DEPARTMENT OF MECHANICAL ENGINEERING SRI RAMDEOBA COLLEGE ENGINEERING - MGMT. NAGPUR - 440 013 3. SHRI RAMDEOBABA COLLEGE OF ENGINEERING AND MGMT. NAGPUR - 440 013 CERTIFICATE This is to confirm that this bonafide recording of the project work called DESIGN ANALYSIS OF UNIVERSAL JOINT SHAFT FOR ROLLING MILLS, PERFORMED ANKIT ANAND (22) M. MUKESH (39) MOHIT KATATE (46) SUGOSH DESHMUKH (68) VASUKARNA YAIN (74) Last year B.E. MECHANIC ENGINEERING in 2013-2014 academic year in partial fulfillment of the bachelor of ENGINEERING degree proposed by RASHANTTRAS TUKAJIDO MAHARAJSHO UNIVERSITY , NAGPUR (Maharashtra) Professor G. R. NIHADE Guide Dr. K. N. Agrawal Dr. S. S. Deshpande H.O.D. Director 4. D E C L L A R T I O N We are a student of the last year of mechanical engineering solemnly state that, the report of the project work called Design Analysis Universal Collaborative Shaft for Rolling Mills is based on our own work conducted during the study led by Professor G. R. Nikhade. We argue that the statements and conclusions made are the result of our research work. We also confirm that the work contained in the Project is original and has been carried out by us under the general guidance of our Guide. ii. We have not reproduced this report or any tangible part of any other literature in violation of the academic thesis. We complied with the standards and guidelines set out in the relevant Decree of the University of RTM Nagpur. Date: PRESENTED, ANKIT ANAND (22) M. MUKESH (39) MOHIT KATHOKE (46) SUGHOSH DESHMUKH (68) VASUKARNA JAIN (74) 5. ACKNOWLEDGEMENTS Apart from the team's efforts, the success of any project depends in large part on the encouragement and guidelines of many others. We took this opportunity to express our gratitude to the people who played an important role in the successful conclusion of this project. First, we would like to thank Dr. W. S. DESHPANDE, Director of RCOEM and Dr. K. N. Agrahal, HOD, Mechanical Engineering Division. We would also like to express our great appreciation to Professor G. R. NIHADE and Professor R. W. PATILE. We can't say thank you enough for their tremendous support and help. We feel motivated and excited the time we attend their meeting. Without their encouragement and guidance, the report would not have materialized. We would also like to thank the technical staff of the Department of Mechanical Engineering. Our thanks are also thanks to Mr. C. S. Pande and Mr. N. M. Dadhe, Sunflag iron and steel limited, Bhandara for their helpful advice and discussion. Special thanks to all Sunflag Iron and Steel Limited staff, Bhandara, for their continued assistance during our project. We want to express our gratitude and love to our parents and siblings, who have been a constant source of support and support. Finally, we thank heaven, from where we always get inspiration and motivation, and with those blessings, we were able to try success in our endeavors. 6. 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Table list: Table No. Page Name 6.1 Speed, R/Min, and Data on The Reduction of 28 6.2 Scope Areas at 31 9.1 Materials of various components 49 10.1 Element quality options 52 11.1 Natural frequency (hyperMesh) 62 11.14 Stressed areas of contact and eye plate 62 10. Chapter 01 UNIVERSAL JOINT 1.1 Introduction of a Universal Compound, also known as Universal Connection, U-Joint, Cardan Joint, Hardy Spicer Joint, or Hook Joint or Compound in a rigid rod that allows the rod to bend in any direction, and is commonly used in mines that transmit rotating movements. It consists of a pair of loops located close to each other, oriented 90 degrees to each other, connected by a cross shaft. The term universal joint was used in the 18th century and was generally used in the 19th century. Edmund Morewood's 1844 patent for a metal coating machine called for universal connection, by that name, in order to accommodate small alignment errors between the engine and the rolling mill shafts. Lardner's 1877 handbook described both simple and double universal compounds, and noted that they were used in the systems of the line shafts of cotton mills. In his treatise on mechanics published in English in 1883, Jules Weisbach described the mathematics of universal joint and dual universal connection. The 19th century use of universal joints encompasses a wide range of applications. Numerous universal compounds were used to link the Northumberland Telescope Control Mines at the University of Cambridge in 1843. For example, the 1881 Ephriam Shay locomotive patent used double universal connections in the locomotive's drive mine. Charles Amidon used a much smaller universal in his bit-brace patented 1884. The spherical, rotary, high-speed steam engine Beauchamp Tower used the adaptation of the universal connection around 1885. Figure 1.1: Universal Joint Mechanical Engineering, RCOEM, RCOEM, 1 11. 1.2 Double Cardan Shaft A configuration, known as the Cardan Joint Drive Double Shaft, partially overcomes the jerky rotation problem. This configuration uses two U-connections connected by an intermediate shaft, with the second U-joint phased in relation to the first U-joint, to undo the angular velocity change. In this configuration, the angular velocity of the controlled shaft will correspond to the speed of the moving shaft, provided that both the moving shaft and the controlled shaft are at equal angles relative to the intermediate shaft (but not necessarily in one plane) and that the two universal connections are 90 degrees outside the phase. This assembly is commonly used in rear-wheel vehicles where it is known as a drive shaft or propeller (propeller) shaft. Even when the drive and control shafts are at equal angles in relation to the intermediate shaft, if these angles are more than zero, the oscillating points apply to the three shafts when they turn. They tend to bend them in the direction perpendicular to the general plane of the shafts. This applies force to support bearings and can cause the launch to shudder in the rear wheeled vehicles. The intermediate shaft will also have a sinusitis component of its angular velocity, which promotes vibration and voltage. Figure 1.3: Telescopic (sliding) Universal Shaft Figure 1.2: Double Cardan Joint Mechanical Engineering Department, RCOEM, Nagpur 2 12. 1.3 The pros and cons of a universal shaft. Universal shafts are the preferred rotating motion transfer device because they are: - Eliminate alignment problems: Flexible shafts do not need the rigid tolerances that solid shafts require - Provide greater design freedom : Limitless opportunities in engine positioning and managed components are more efficient: Flexible shafts are 85%-95% efficient. Gears, U-Joints Belts and Pulleys give much lower performance due to large friction losses - light weight and powerful: flexible shafts have a good weight advantage compared to other design solutions when transferring large loads of power - have a lower installation cost: Flexible shafts are installed in minutes without special tools or skills. Solid trees, Gears, Pulleys and universal connections require fine alignment and skilled mechanics for its installations. Reducing the cost of spare parts: Bearings and hulls for hard shafts and gears require accurate processing. Flexible shafts eliminate the need for such demanding tolerances and their excessive costs. Easy to install: you don't need special installation tools. Can be developed in the latter stages of the project: Unlike other rotating motion devices that need to be worked around because of their rigidity, defined configurations and large Flexible shafts allow greater freedom of design because engineers have only one part to work on, eliminating complex coordination Several Parts 1.4 Flawed Cardan Joint suffers from one serious problem: even when the shaft entering the shaft rotates at a constant speed, the exit of the disk shaft of the axis rotates at variable speed, thus causing vibration and wear. Speed and acceleration increases from an operational angle. Other drawbacks are: grease is required to reduce wear. Shafts should lie in exactly the same plane. A reaction that's hard to control. Department of Mechanical Engineering, RCOEM, Nagpur 3 13. 1.5 Applications - Airplanes - Thrust Reverser Drive System - Variable Bleeding Valve - Drive Valve - Car Pedal - Pedal-Adjustable Door - Power Sliding Door - Power Seat Track - Tilt Steering Industrial - Road Spike - Power Sander - Power Tools Medical - Breast Delivery - Corn Flapeal, RCOEM, 14. CHAPTER 02 LITERATURE REVIEW To understand the scope of the project, a literature review is a must. The literature review for this project is as follows. 2.1 Introduction of Robert Hook is usually regarded as the inventor of 'Hooke's joint' or 'universal joint'. However, it is shown that this flexible compound (based on four armed cross rotates between the semicircular igo attached to the two mines) was actually known long before the time of hook, but was always supposed to give an output exactly consistent with the entrance shaft. The hook carefully measured the relative movements of the two axes and found that if one of them is inclined to the other, the even rotation of the input produces a different rotation speed. Huque's research on the universal joint led to him being identified with his name, and he ended up far more important as a rotating bond than as a solar counterpart. The more complex versions later developed by Hook included a provision for two main connections to be connected by an intermediate shaft. If the phase and shaft angle are set up properly, this double hook joint can reverse the variable output rate typical of a single all-rounder. It proved to be invaluable for modern car transmissions. 2.2 Corner connections Before you continue with this device you need to look back to an earlier phase in the history of the mechanisms. The need to transfer the turning motion from the primary shaft to the second shaft at an angle to the first (rather than directly into the line or parallel to it must have arisen repeatedly from ancient times, and was solved by a number of anonymous artisans. in 1664, that angular drive could have been achieved using beveled gears, but was more easily achieved with a chain of devices individually known as paradoxum. He illustrates this mechanism with woodcuts lending to his unpublished manuscript of Chronometria Mechanica Nova by a deceased anonymous author identified only as Amicus. The paradox can be seen to be made up of the same two forks associated with the four armed cross that characterize Hook universal. Department of Mechanical Engineering, RCOEM, Nagpur 5 15. 2.3 The main instrument of the 1667 universal Hook joint was too light and rough to function satisfactorily as a flexible compound to transmit significant torque at an angle; although he says it can facilitate the operation of the wheel and other mechanical uses. It was a study of the Hook movement of the universal joint, and its propaganda and application mechanism that led to it becoming commonly known as the Hook Joint for millwrights-at least in England. 2.4 Huh's mathematical analysis was a competent mathematician, but the spherical trigonometry of his time was not sufficiently developed to allow tan and tanning, because the analysis of the kinematics of the universal joint. Only in 1845 has Poncelet proved that where the α is the corner position of the entrance shaft, β is the angular position of the output shaft, μ is the tilt of one shaft to another - the corner of articulation. 2.5 Practical testing 2.5.1 Models of working models of Hook joints were built to illustrate both the cross and disc shape of the inner penis. They are shown in (Figure 2.2), along with commercial examples of steel and acetal plastic. Figure 2.2: The Single Hook Joints. (Top to bottom) with a cruciate member of the interior; With a disco-shaped member of the interior; two commercial versions. Figure 2.1: Universal Joint (paradoxum) Schott (1664) Department of Mechanical Engineering, RCOEM, Nagpur 6 16. 2.5.2 A test rig was built to study the compound at different angles of articulation shown in (Figure 2.3). Originally it was used to find the maximum angle of articulation allowed by the various designs shown in the picture above. A 90-degree slope will obviously not be possible as a result of geometric limitations, but smaller angles are limited by mechanical design when the tip of one fork comes into contact with the base of another near its shaft. Figure 2.3: A device to study the movement of the joints of one or double hook at different angles of articulation. 2.6 Hooke 2.6.1 Single Compound Connections, but with adjustable Hooke arms the first proposed application of the basic universal all-metal was to provide a drive in azimuth (from the polar axis) to the astronomical quadrant set in such a way that its own Manual height adjustment (Figure 2.5) is allowed. A simple universal wouldn't work in this situation. Figure 2.4: Hook's proposed azimuth montage for quadrant (1674). Figure 2.5: Hook design for universal with infinitely adjustable hands. Department of Mechanical Engineering, RCOEM, Nagpur 7 17. 2.6.2 Double Hook itself realized that there may be circumstances in which periodic velocity fluctuations that characterize the standard form of the compound will be detrimental. His decision was to use two all-rounders, one at each end of the overall intermediate shaft, to give a double joint (Figure 2.6). The units can be assembled in stages, 90 degrees outside the phase or in any intermediate position. The orientation that leads to the transmission of movement 90 degrees outside the input phase, combined with parallel input and exit shafts (or equal angles of inclination to the intermediate shaft) negates the change, and the entire assembly is now widely known as the constant speed of the connection. They are available commercially in a variety of sizes. Adding a splicing of the assembly to the intermediate shaft allows for back and forth movement and provided a connection that, centuries after Hook's time, proved extremely successful in the car. This allows energy to be transferred from the engine to the wheels, even if their relative position changes as a result of the over-stroke and troughs on the road. 2.7 Shaft failure modes can fail by: - Excessive lateral deviation, which causes items such as gears to move sideways from their proper position, resulting in the wrong grid. - a torsion deviation that destroys precise angular relationships or timing between sections of the mechanism. Wearing can occur on carrier surfaces (magazines) or other contact areas such as cameras. Fracture. If the shaft has been grossly underdeveloped, the fracture usually occurs due to cracking fatigue. This is stated below. Figure 2.6: Double Hook Joint. Department of Mechanical Engineering, RCOEM, Nagpur 8 18. 2.7.1 The mechanism of fatigue cracking of fatigue cracking occurs in conditions of re-loading or cyclical load. The failure is not the sheer amount of load. This is the cumulative effect of many thousands, often millions, of repetitions of loading cycles or loads that cause cumulative damage. The bearing ensures that as the shaft rotates, the mass media always hangs vertically down. It should be clear that the force due to the medium mass f th mg. This force applies the moment of bending to the right side of the shaft and the moment of bending due to F reaches the maximum on the bearing shaft. Consider point A at the top of the shaft. Since the F force tends to bend the shaft down, the material of the shaft on the A should be in suspense. Similarly The material at point B at the bottom of the shaft should be loaded into compression. Now let the shaft rotate through 180o, so point A moves to the bottom and B to the top. The material of the shaft in A is currently in compression, and the material in B is in suspense. After further rotation of the shaft 180o, A is again at the top and in suspense, while B is again at the bottom and is in contraction. A total of 360o shaft rotation results in one full load cycle for points A and B and, in fact, for all points on the shaft. The figure shows a graph of load cycles for points A and B. Figure 2.7: Cracking Fatigue Phenomenon Department of Mechanical Engineering, RCOEM, Nagpur 9 19. 2.7.2 Spreading crack fatigue within certain limits, applying a large number of load cycles can cause tiny cracks to occur on highly stressful parts of the component. Subsequent load cycles cause this microscopic crack to increase in size, slowly at first, but with increased speed. If not detected or tested in any way, the crack will eventually become so large and will so weaken the component that it will fracture completely. The final failure can be sudden and can be catastrophic; however, cracking fatigue is a progressive failure that occurs for some time before the final failure. Because of the insidious nature of cracking fatigue and its potential to cause catastrophic failure, designers should always be alert to design features that can lead to fatigue failure. Items such as shafts that can go through a very large number of rotations during their lifetime should always be designed with fatigue failures in mind. 2.7.3 Concentrations of stress In practice, it is observed that fatigue cracks often occur where there is a sharp or sudden change in the shape of the component. Theoretical analysis and experimental methods show that the stresses inside the material increase dramatically with sudden changes in the section. These functions are called concentration of stress or increased stress. Referring again to the aforementioned figure, it is clear that there is a sharp change in the diameter of the shaft next to the bearing 2. This diameter change is described as a shoulder on a shaft. Such a shoulder causes a significant concentration of stress. The location of the shoulder is already in the area of high moment bend and therefore high tense and squeezing stresses. The combination of these two factors makes the cross-section of the shaft particularly vulnerable to fatigue. Note that in the picture above, the maximum moment of bending on the shaft actually occurs on bearing 2, but the smaller diameter of the shaft and the concentration of stress due to the shoulder makes this a section at which fatigue failure will occur. Cross-section changes other than shoulders that cause effects stress in the mines, include: grooves - Threads - Oil Holes - Mechanical Engineering Department, RCOEM, Nagpur 10 20. 2.8 Principles of DesignIng The Role of Joints and Drive Shafts is to transfer torque between shafts that are not in the line. These transferable loads are limited by the capacity of the materials used. Therefore, when assessing the capacity of the joints, it is most important to determine the pressure between the mobility and their caterpillars. However, it is necessary to decide whether short-term or long-term loading is related: - For short-term loading, stress is quasi-static. The connection is then designed so that excessive plastic deformation does not occur, - For the long-term load the force position is dynamic. The joint should be designed for many millions of stress cycles. Longevity can only be determined by extensive drilling tests, and purely mathematical approaches are still insufficient. Only in the case of Hook joints can life be calculated using modified methods from the practice of roller bearing and Fisher's empirical equation. This cannot yet be used for ball and pod joints. Joint manufacturers use formulas based on the empirical Palmgren Eq. (2.1) to transform durability from drilling tests into a formula that suits other torque, speed and angle conditions. When designing a technique, the designer does not know all the parameters from the beginning, so he or she needs to design on the basic data, some of which are evaluated. The designer then overestimates the joints with precise values for static and dynamic stresses and can choose a connection from the manufacturers' catalogs: At this stage, all other factors such as maximum angle articulation, plunge and installation capabilities should be taken into account. If we consider P1 and C as a reference load at which, at a constant rate, the moving pair reaches life L1 and 106, the indices can be omitted and the equation is written as L No. 106 (C/P)p. (2.1) Figure 2.8: Life vs. Unsuccessful Bearing Numbers of the Department of Mechanical Engineering, RCOEM, Nagpur 11 21. 2.9 Hook connections and joint drive shafts of the Hook Torque The capacity of the joints is determined by the position of the moving surface in relation to the rotation axis z. In the case of hook connection, the P load acts at a pressure angle of 90 to the axis and at a sloping angle of g y 0 to the rotation axis z (Figure 2.9). This is due to the fact that the p perpendicular line of action z.Each transmitting element consists of a number of movable bodies. 2.10 Universal Collaborative Projects: Cardan joints for industrial use consist of variations using a cross and bearing design. All designs (except for simple or composite bearing types) transmit torque through anti-friction bearings; two fake or cast igs and cross a member. Heavy lip seals that are subject to an eccentric boot prevent the leak of lubricant and and This allows the atmosphere to be exploited, for example, in foundry applications. Since crowned videos are commonly used, U-sharing can be statistically defined. Solidified high-precision race tracks with bearing steel rollers are used for optimal life. Figure 2.9: Contact Persons Position relative to rotation axis on trunion, a. side view, b. plan view, Department of Mechanical Engineering, RCOEM, Nagpur 12 22. Mill universal joints can be classified as one of the following five types: - Closed Eye (1-piece) needle design (Figure 2.10) - 1-piece of thethah surrounds bearing housing. Split igo design (Figure 2.11) - Yokes separated the axis to produce a 1-piece bearing dwelling with a solid bearing end of the lid, whose iga halves are held together with a bolt tie during assembly, delivery and installation. After assembling the mill, the halves are brought together by bolts of communication with the flank of the drive shaft. Split Eye Carrier (Figure 2.12) - Bearing wells are divided into two sections and retained by a combination of serratium and bolts. Type block (Figure 2.13) - Bearing enclosures bolted to yokes with keys from the face, a one-part bearing shelter bore. Composite simple bearing - Only a design without roller bearings. The bearings are usually made of non-ferological material, such as self-glazing composite (used at low speed, heavily contaminated, high-temperature applications such as steel continuous slab casting machines. 1-piece yokes have the advantage of not maintaining a requirement to check if the bolted compounds are loosened and eliminate the accuracy of matching components for an equal figure 2.11: split yokeFigure 2.10: Closed eye 1 piece and Figure 2.13 : typeFigure 2.12 block: split bearing eyes. Department of Mechanical Engineering, RCOEM, Nagpur 13 23. bearing the load. Corrosion issues associated with split-bearing eye designs are also avoided. The 2-piece yokes and block-type designs were designed to increase the diameter of the transverse trunnion and the torque-building U-connection, allowing the assembly. Split-yog designs are unique because the tie bolt is not used to transmit torque or hold the yoke components after installation. All structures have the ability to splice the central sections to ensure that the length is compensated for several changes during operation. Splines can be hardened, usually nitriding, for applications with frequent travel ases. The central sections can be pipe welded directly to yokes or flanged to provide a more economical subring of sparing U-joint parts. It is rare for an integrated facial pads machine or special spline teeth between the flanks (Figure 9 and 10) on high load applications. 2.11 General considerations in the design of the tree. 2.11.1 To minimize and stresses, the length of the shaft should be as short as possible and and Minimize. 1.11.2. Cantilever beam will have more deviation than just supported (straddle installed) one for the same length, load and cross-section, so cross-border mounting should be used if the cantilever shaft is dictated by design limitations. 1.11.3. The hollow shaft has a better stiffness/mass ratio (specific stiffness) and higher natural frequencies than a relatively hard or strong hard shaft, but will be more expensive and larger in diameter. Figure 2.15: Radial Face Line. Figure 2.14: Integral Tinctures of the Mechanical Engineering Department, RCOEM, Nagpur 14 24. 2.11.4. Try to find stress-raisers from regions of great moment bend, if possible, and minimize their effects with generous radii and relief. 2.11.5. Total low-carbon steel is as good as higher steel strength (because deviation is a typical design limitation issue). 2.11.6. Deviations on the gears carried out on the shaft should not exceed about 0.005 inches, and the relative slope between the aus should be less than 0.03 degrees. 2.11.7. When using simple bearings (sleeves), the deviation of the shaft along the entire length of the bearing should be less than the thickness of the oil film in the bearing. 2.11.8. When using bearings of non-self-leveling elements, the slope of the shaft on the bearings should be below 0.04 degrees. 2.11.9. When there are loads on the weight thrust, they should be hung on the ground through one bearing thrust in the direction of the load. Do not divide the loads between the traction bearings, as the thermal extension of the shaft can overload the bearings. 2.11.10. The first natural frequency of the shaft should be at least three times higher than the expected operating frequency of exposure and, preferably, much higher. (The preferred factor is ten or more times, but it is often difficult to achieve). 2.12 Design considerations are taken for the shaft For the design of the shaft the following two methods - Strength-based design: In this method, the design is carried out in such a way that stress anywhere in the shaft does not exceed the material stress of profitability. However, is not considered for the deviation of the shaft and the shaft twist is included. Design based on stiffness: The basic design idea in this case depends on the permissible deviation and rotation of the shaft. The load on the shaft at any time depends on the nature of the load, which works on it. The stresses that may be present are the following. Basic stress equations: σ b' 32M 2d0 3(1'k4) Where, M: Bending moment at point of interest do: Outer diameter of the shaft k: Ratio of the internal to the outer diameter of the shaft (k No. 0 for a solid shaft, because the internal diameter is zero) Department of Mechanical Engineering, RCOEM, Nagpur 15 25. - Axial Stress σ a 4'F o 3 (1 q2) Where, F: Axial force (tense or compressed) α: Column Action Factor (No 1.0 for download) α was put into the equation. This is known as the column action factor. The factor of the column is caused by the phenomenon of the buckle of long slender members, which are affected by the snous compressed loads. Here, α is defined as, α No. 1 1 - 0.0044 L/K for L K qlt; 115 , α y σ yc q2 nE (L K)2 for L K qlt; 115 Where, n No. 1.0 for hinged end n No. 2.25 for fixed end n 1.6 for partially restrained purposes, as in bearing K - the smallest radius of gyration, L - the length of the syc shaft - stress output at compression - Stress due to xy xy xy 16T π d o 3 (1 k4) Where, T: Torque on the shaft sxy: Cher stress due to xercyon combined bending and axial stresses and bending and axial stresses are normal stresses; hence, pure normal stress is given, the following equation. Pure normal stress can be positive or negative. Typically, the stress of haircuts due to xersion is only seen in the mine and haircut stress due to the load on the shaft is neglected. - Maximum stress theory σ b' 32M x 0 3 (1'k4) - 4 D o 3 (1'k2) Department of Mechanical Engineering, RCOEM, Nagpur 16 26. The design of the shaft mainly uses the maximum theory of haircut stress. It says that the penis of the machine fails when the maximum stress of the haircut at the point exceeds the maximum allowable stress of removing from the shaft of the material. Thus, the aforementioned equation could be replaced with the values ps and hksi (x 2) Thus, the final form is acceptable 16 yd o 3 (1 k4) M. Fd o(1 q2) 8 2 t2 the diameter of the shaft can be calculated in terms of external loads and properties of the material. However, the above equation is additionally standardized for steel valing in terms of acceptable construction load factors and load factors in the ASME design code for the shaft. - ASME design code shafts tend to operate on gradual and sudden loads. Thus, the equation changes in the ASME code with the help of suitable load factors, 16 yd o 3 (1 k4) C bm M and fd o (1 q2) 8 2 (Ct T)2 2.13 Joint Selection (Torque Rating) The capacity of the universal connection is the torque that can transfer the joint. For this joint, it is a function of speed, angle of work and conditions of service. Section 2.13.1 shows usage factors based on speed and working angles for two maintenance conditions: intermittent operation (say, operation or less than 15 minutes, usually regulated by the need to dissipate heat) and continuous operation. The torque power of a single Cardinal connection to a standard steel structure is determined as follows: from the required speed in The angle of work in degrees and maintenance status (intermittent or continuous) find the appropriate usage factor from the table. Multiply the torque you need to pass on to the shaft's input using the utilization factor. If the app includes a significant amount of shock load, multiply by an additional dynamic factor 2. The result should be less than the static tipping point of the joint. Department of Mechanical Engineering, RCOEM, Nagpur 17 27. 2.13.1Factors for the torque rating of universal joints 2.14 Selection criteria 2.14.1 Torque requirements: The first step in choosing the Cardan shaft is to determine the maximum torque for transmission. Maximum torque should take into account the maximum torque of the main engine, including the inertial effects of the prime run when slowing down during overload. The prime torque of the engine is usually seen as unevenly split through the Pinion stand in the range of 40 to 67% to provide unequal torque to boot mill rolls. This application torque is adjusted by applying the appropriate service factor (s) in accordance with the joint manufacturer's recommendation. The resulting choice of torque is compared to any: endurance U-connection or fatigue torque rating for reversing applications; or one way or a pulsating fatigue torque rating for non-reversal applications. Both rankings are based on the material strength of the shaft. One way torque rating is usually 1.5 times to reverse the fatigue torque rating. 2.14.2 Maximum bending moment During joint operation the moment of bending occurs in the connecting shaft in the function of the working angle of incorrect action and torque of movement. This moment of bending causes a load on the supporting bearing, which periodically with two full cycles on the back of the shaft. The moment of bending is always in the plane of the yoke's ears. The maximum bending moment is given to M and T tan (α) Where, M and maximum bending of the Department of Mechanical Engineering, RCOEM, Nagpur 18 28. T - operating torque α - operating angle 2.14.3 Axial travel requirements and downforce Gear spindles are designed to accommodate angular delays and small changes in the length of the travel axis, having their hubs transverse inside or pull in sleeves. However, U joints must have a clearance fit in connected equipment or a spline travel section in an intermediate shaft. A clearance or slip fit allows the end roll to slip out or retractable under unpreparedness. The amount of retractable is calculated from: Pull out: Pull out of THE CL (1- cos) CL - center to the center line of the joint flexible location α - operating angle of Axial's blunder alignment You joint spline under torque leads to axial force applied to support bearing these forces are the friction factor function of the operating torque of the operating angle and the spline diameter step axial force can be calculated calculated Facsimile - 2T'cos Pd Where, T - Operational Torque U - Friction factor α - operating angle Pd - spline pitch diameter Same equation is used for spindle gear strength with the length of a compensating spline. For spindles without length compensating splines the same equation is used expect the Pd equally to the topped gearing used. The larger step diameter will reduce the a heavy force proportional to the Department of Mechanical Engineering, RCOEM, Nagpur 19 29. 2.15 Special Consideration 2.15.1 The applicability of Universal Connections is particularly good for angular reversal requirements for coarser mill applications. The rolls of these mills are usually powered by two separate direct drive engines that are synchronized by the mill control system. Mills should run through a relatively large range of angular. The range of 5 to 2 per graph is common. In these applications, it is not recommended to use an englishness exceeding 6 degrees. Because of the engine overload capabilities built into these mills and the low minimum speed of the engines, it is more appropriate to consider the density of the application's torque rather than the density of power. The motors for this application usually have a current limit set between 220 and 250% of the engine plate rating. When congestion factors and minimum engine speeds are taken into account, the designer may encounter an 8,000 or 10,000 hp mill engine that can produce 1.5 million feet of torque at the current limit and is capable of generating torque spikes in excess of 2.0 million feet-pound. This torque should be transmitted through a universal joint spindle well suited to these applications. Compared to the bench finish, these types of mills work relatively slowly. The faster the stacked mill rarely exceeds 120 or 140 rpm respectively, the dynamic balancing of the spindle component is usually required. 2.14.2 Choosing the size of the choice for reverse mills requires careful consideration of all operating conditions. Because of the congestion capabilities, using a torque directory, ratings are generally not useful and can be misleading. The two main factors that usually determine the final size of the U-connection are the minimum diameter of the work roll; and who clarifies the U-joint. To get the maximum possible reduction per pass, mill designers (especially in steckel plants) define working rolls as small as possible in diameter. The question of how to get the necessary torque to work rolls is usually a secondary consideration. As a result, the diameter of the reset rolls work in many cases is a deciding factor when choosing the size of the versatile joints, at least for the roll end of the U-connection. Often Spindel finds the department of mechanical engineering, RCOEM, Nagpur 20 30. that universal joints, that is, for example, 900 mm dia. You may need to meet the desired congestion criteria. However, the size of the reset rolls work for the mill is 850 mm. The diameter of the roll joint swing should be less than 850 mm. This does not have a negative impact on the functionality of the mill. This only means that the life of the universal joint bearing will be shortened. Department of Mechanical Engineering, RCOEM, Nagpur 21 31. Chapter 03 PROBLEM IDENTIFICATION AND APPROACH 3.1 Sunflag: Details of the BSM Detailed study of the process in Sunflag Iron and Steel Limited, Bhandara was conducted. The main focus was the bar and the mill section. The department is equipped with 60 tons per hour, 20 booth, 2 high continuous Mannesmann designed rolling plant, duel fuel walking hearth-like re-heating furnace, management process through ABB automation, online evaluation system, closed television chain system for harvesting and bar movement. 3.2 Problems Revealed Rolling Mill stands are controlled by an electric motor, power depending on the number of the stand. Various problems associated with the installation of a rolling plant are excessive shock, failure of bearing, failure of the shaft, etc. Failure prone areas of the shaft are needle bearings of a universal joint cross or spider and yoke pin or detection of contact failure. The spider has needle bearings, which fail almost every 3 months, damaging the needle pin. The Igo pin also faces severe stress in crushing, bending and wear out frequently. 3.3 The actual problem of the blank has a large cross-section area at the entrance to the stand. It strikes the video, causing a jerk. In addition, the blanks have some small air pockets trapped inside them. When the blanks pass between the rollers, these air pockets are compressed and its temperature increases simultaneously. As a result, they explode in small proportions. This causes reactionary force on rollers, normally to its axis. Some of the jerk is produced because of this being passed together through the wobbler's ifo pin. This causes the ifth to wear out the area between its surface and the surface of the eye plate. This increases the gap between their surfaces and eventually, the shaft begins to fluctuate more. This, in turn, leads to the fact that the bearing can not be crushed. If there is no relative movement between the components in the universal joint, the assembly finally gets stuck. 3.4 The approach followed by the project is: Exploring the rolling process in the bar and sectional sectional plant at SUNFLAG IRON AND STEEL, Bhandara. Department of Mechanical Engineering, RCOEM, Nagpur 22 32. Study of the various failures occurring in the mine. Understanding the causes of the failure and how to repair the mines. Acquisition of standard drawings and other information, such as shaft material, frequency necessary for the rolling process. Calculating the torque of the design for the shaft design. Mine simulation at SolidWorks 2013. Pre-processing of the shaft in HyperMesh11.0.0.39 - Solution in National. Post-processing in HyperView. Getting results such as o Deflection, o Stress induced in various sections, o Critical sections, o Reactions on bearings - Offering design changes to reduce failures and increase the time between repairs. Department of Mechanical Engineering, RCOEM, Nagpur 23 33. Chapter 04 PROCESS AT BSM, SUNFLAG STEEL, BHANDARA - Rolling is done in 5 stages through variable reduction process: o 1-4 - Roughing Mill o 5-8 - Roughing Mill o 9-14 - Intermediate Mill o 15, 17, 19 - Finishing mill o 16, 18, 20 - Vertical undersea - A total of 23 rental machines are located in the line to reduce the size of the blank (130 x 130 mm sq.) from rough to finishing (5.5 mm) stage. The different types of mills used in the steelworks are described below: 4.1 Roughing Mills: There are four pairs of shafts, i.e. 8 shafts used to provide electricity to rough mills. The shafts are measured from 1 to 8 and are divided into two sections. One of them consists of shafts from 1 to 4, and the other from 5 to 8. Rough mills quickly remove a large amount of material. This type of mill uses a wavy tooth shape carved on the periphery. These wavy teeth form many successive cutting edges produced by many small chips, resulting in a relatively rough finish surface. During cutting, several teeth are in contact with the work piece reducing chatter and vibration. Rapid removal of stocks with heavy milling cuts is sometimes referred to as hogging. Rough mills are also sometimes known as copy cutters. A greater reduction in diameter occurs in this mill, hence the stresses and strength at the transmission mine are the highest here. 4.2 Intermediate Mills: There are six shafts used to transfer energy to intermediate rolling mills. The trees are measured from 9 to 14. The thickness of the blank decreases further, but not by the same margin as at the rough mills. The thickness after exits from the mills is close to the required cost. The shaft associated with this mill was chosen for study and analysis in our project. Department of Mechanical Engineering, RCOEM, Nagpur 24 34. 4.3 Finishing Mills: There are three mines used to transfer energy to finishing mills. The promers trees are 15, 17 and 19 respectively. In this type of mill, the mill is finished to the required size and shape. By now, steel has been rolled into a flat bar as long as 200 feet. Unlike rough mills, the finishing mills roll the transfer bar in tandem, meaning each bar will roll through all six stands at the same time. Hot steel is quite fragile as it is rolled and the tension between the finishing mills must be carefully monitored at very low levels in order to avoid stretching or breaking the strip. Until the rolling operation is complete, the head and tail-ends of the transfer bar will be a haircut to square them, helping to ensure proper threading and tail-out. The final two-cup descaling operation is performed to clean off the scale that grew onto the bar during the rough. It is important to smoothly adjust the gaps and roll speeds to maintain a stable rolling strip to the required thickness, despite the temperature fluctuations present in each bar. 4.4 Vertical beating - The vertical stand of the rolling plant is a combination of horizontal and vertical mill stand, which is used where the speed of the material is very high and also we can roll the material without the use of a twist pipe. To reduce the slab to 150mm wide and form a blank edge, this vertical rolling mill stand is equipped with rolls with grooves and hydraulic screws down the mechanisms. It has a heavy design, a strong design, requires low maintenance, has easy operation and is dimensionally accurate. Department of Mechanical Engineering, RCOEM, Nagpur 25 35. CHAPTER 05 ROLLING MACHINE Parts of a mobile car 1. Roller guides with roller bearings 2. Shafta transmission 3. Universal connections with wobblers 4. Transmission 5. Flang 6. Motor 7. Rollers 1. Roller guides with roller bearings: Roller guides are guides used to ensure that the rollers rotate in the right direction and with less friction between rollers and bearings. Roller bearings are bearings that support rollers when they turn. Rollers should be able to withstand enormous forces and therefore guides and bearings play an important role. 2. Transmission shafts: This is a mechanical component for torque and rotation transmission, commonly used to connect other drive train components that cannot be connected directly due to distance or the need to provide relative movement between them. Transmission shafts are carriers of torque: they are susceptible to xersion and haircut stress, which is equivalent to the difference between input torque and load. Therefore, they should be strong enough to withstand stress while avoiding too much extra weight, as this, in turn, will increase their inertia. To allow for changes in alignment and distance between driving and guided components, transmission trees frequently include one or more universal joints, jaw joints, or rag joints and sometimes splined link or prismatic connection. Figure5.2: Rolling Mill.Figure 5.1: Val and Installation Roller by Department of Mechanical Engineering, RCOEM, Nagpur 26 36. 3. Universal With wobblers: The compound is in a hard rod, allowing the rod to bend in any direction, and is commonly used in mines that transmit a turning motion. It consists of a pair of loops located together, oriented to 90 to each other, connected by a cross shaft. Wobblers prevent the oscillation of the shaft, which is an unstable movement from side to side. This ensures that the shaft rotates properly and therefore transfers energy to rolling mills. 4. Gearbox: The gearbox is a mechanical device used to increase the torque of the output or change the speed (RPM) of the engine. The engine shaft is attached to one end of the gearbox and through the internal transmission configuration of the gearbox, providing this output of torque and speed determined by the ratio of transmissions. 5. Flank: a rib or rim for strength, for guidance, or for attachment to another object. It is located after the gearbox, but in front of the engine. It ensures that the engine and gearbox are properly attached as this is an important factor for the transmission of electricity between the engine and the shaft. 6. Motor: It provides power for a complete rolling process. It converts the generated electricity into mechanical power and transfers it to the mine through a transmission transmission at the required torque and rpm 7. Rollers: Rollers are the most important part of rolling cars. They carry the blows of blanks and squeeze the working piece on a new cross-section. Department of Mechanical Engineering, RCOEM, Nagpur 27 37. Chapter 06 BASIC CALCULATIONS AND ASSUMPTIONS 6.1 Data from the Sunflag Iron and Steel plant (actual time to work time) Table 6.1: Speed, R/Min and Reduction Data. Ref stand speed. rpm Actual rpm decrease 9 1.32 720 719 1.35 10 1.72 647 647 1.31 1 1 2.25 7 The above data is obtained for the final section of RND 26 (i.e. Round with a diameter of 26 mm). The length of the harvest is 6,652 m Expected to be finished - 400 m Billet (raw) class SAE 1020 a. Clutch data: Material: EN 24 i.e. SAE 4340 Motor rpm: 720 rpm Transmission ratio: 9.109 Ent rpm Roll data: Roll diameter Do 360.5 mm Roll groove height 12.72 mm Roll radius external and 180.25 mm Roll internal radius Temperature data: The harvest temperature when leaving the furnace No. 1200 oC The temperature of the blanks at the stands Nos. 9 and 950-1000 oC Environment temperature 30-40 oC Department of Mechanical Engineering , RCOEM, Nagpur 28 38. 6.2 The basic assumption of a rental installation is being considered for this project to bar a mill for hot steel rolling. Creating a closed expression of the form for the torque of the roll in the rod rental is very difficult. The guiding differential equation is much more complex than in flat rolled, as the state of stress can no longer be considered as two-dimensional. The geometry of the roll aisle is more complex to describe, and all other aspects such as the description of the amount of heterogeneous deformation, interfacial etc. are also present in rolling rods. The traditional approach is to or an oval bar in an equivalent square section, and then use some analytical expressions for flat rolling. Since the expressions of roll torque in flat rolled are not very accurate, you can't expect these expressions, combined with equivalent rectangle methods, to work well. Another approach to try to take flat expressions rolling into the rolling rod was introduced by Y. Lee and Y H Kim. In their bank power study they introduce a weak plane-voltage concept to produce a closed form expression for bank power. Since the tension in the side direction in the rolling of the web is limited by the radius of Lee's groove and Kim argues that the position of force can be considered to be in the weak condition of the plane. They introduce a simple linear function to compensate for the three-dimensional state of voltage in the roll gap in the roll roll. The pressure in the rod rolling is then presented as: p rod No (1 - ε 1)p plane strain Where, No. 1 is a lateral deformation. The equivalent rectangle method is used to calculate the average effective voltage and voltage rate. They assess their expression of measured values from one round to the oval and from one oval to the round passage at different temperatures. The average difference between the measured and calculated roll strength at three different temperature levels was 0.8-5.7 percent, where calculated values overestimate the strength of the roll. Since the experiments were conducted in one round before the oval passage and one oval to the round aisle, no conclusions about the consistency of the model in the geometry of the data roll pass can not be made. They do not consider the calculation of roll torque, and therefore it is noted only that the weak concept of plane deformation can be a fruitful way to find a consistent model of roll torque in round oval sequences. This method is based on the use of statistical design and modeling of FEM modeling results. The equivalent rectangle is a rectangle with the same area as the circle, but the cross-section is a square rather than an original section. Department of Mechanical Engineering, RCOEM, Nagpur 29 39. 6.3 Calculating the cross-section area at the entrance and exit, to stand No. 9: Initial section area no 160 × 160 mm2 - 25600 mm2 Length - 8 meters Final round diameter section - 26 mm Area No. 2 4 530.93 mm2 To calculate the final length we have the equation A1L1 2L2 25300 X 8000 and 530.93 X L2 L2 385.74 m At the first pass , the cross-section area remains the same, as the reduction rate is 1.00 (the reduction rate of 1.00 means zero reduction of the area). The cross-section varies from square (initial cross-section of blanks) to oval. Such The area of the cross-section of the circular harvest after passage Nos. 1 and 25600mm2 When passing 2 the reduction factor is which means a 30% reduction in the area therefore the area of blanks at the exit from the pass No. 2 No. 1 - 30 100 × 25600 and 17920mm2 Department of Mechanical Engineering, RCOEM, Nagpur 30 40. Thus, the cross-section area after each pass is displayed in the following table: Table 6.2: The prep areas at the entrance and exit of the aisle. Side blanks 160 mm standno. Rfactor Areaatinlet Areaatexit 1.00 1.000 25600.00 25600.00 2.00 1.300 25600.00 17920.00 3.00 1.290 17920.00 12723.20 4.00 1.270 12723.20 9287.94 5.00 1.280 9287.94 6687.31 6.00 1.210 6687.31 5282.98 7.00 1.270 5282.98 3856.57 8.00 1.280 3856.57 2776.73 9.00 1.350 2776.73 1804.88 10.00 1.310 1804.88 1245.36 11.00 1.310 1245.36 859.30 12.00 1.280 859.30 618.70 13.00 1.000 618.70 618.70 14.00 1.000 618.70 618.70 (Highlighted rows indicate that the passes are shutoff due to process requirement) In the above calculations , the section area at the exit of one passage is considered to be equal to the area of the cross-section at the entrance to the next passage (i.e. the effect of drawing is ignored) The blank is simultaneously located in several stands in the rolling mill. Rollers pull blanks with different forces. This leads to the effect of drawing in the blank. This area is shrinking due to inter-mail. Here, for the ease of calculation, this effect is ignored. Because the main objective of the analytical project is to consider the failure of the universal joint assembly of the shaft, the inter-supply effect, which reduces the torque used, is ignored. This raises serious concerns when rollers or bearings need to be analyzed or developed by the Department of Mechanical Engineering, RCOEM, Nagpur 31 41. Thus, the area of blanks at the entrance to the booth No. 9 i.e. A1 No 2776.73 mm2 Cross-section area when exiting the stand No. 9 A2 1804.88 mm2 As the blank passes through the stands its cross section after passing from each stand is different from the shape with which it feeds in the rollers i.e. for a specific stand, if the output of the cross section is a round for the next entry cross section becomes an elliptical section that means that if the input section is round, the exit section should be elliptical and vice versa. Thus, the diameters of the blank in the ins and exits section are calculated as follows the blank diameter at the entrance of D1 2776.73 × 4 × 1 π and 59.46 mm Similarly, Diameter at the exit: D2 and 47.94 mm (for perfectly round shape) But, since the section at the exit is not fully circular and elliptical due to the geometry of the roll, 1804.88 and π × hR 2× Rw2 Where , h2 and 17.67 mm that is from the geometry of the roll and roll gap given the ∴ w2 and 32.6 mm Figure 6.1: the incoming and outgoing geometry section works the piece on the roll aisle. (a): Oval circle, (b): Oval Circle department of the Department of Mechanical Engineering, RCOEM, Nagpur 32 42. In accordance with the above figure and in accordance with the above figure the height and width of the working part (i.e. 2h and 2w) equal to the diameter of the working part Thus, 2w1 - 2h1 - D1 - 59.46 mm w1 , D1 2 - 29.73 mm, and h1 - D1 2 - 29.73 h2 - 17.67 mm and w2 - 32.6 mm -----------A from the geometry of the roll and the area. Figure 6.2: Actual geometry of roller blanks. Department of Mechanical Engineering, RCOEM, Nagpur 33 43. CHAPTER 07 ANALYSIS FOR CALCULATION FOR ROLL FORCE AND TOR'UE 7.1 Contact length or roll bite length: This is the distance between two dots on the rolls, along the circle at which the working piece enters and leaves the rollers. In this case, L r No 2Ri (h1'h2) Ri - Inner roll radius - 360.5-2∗12.72 2 - 167.53 mm, replacing Ri, hi 29.73 mm, h2 and 32.6 mm, L r' 63.56mm For the equivalent assumption of a rectangle, the height of the section 2h at any distance x from the entrance is given h h1 - L r Ri x x ∗2 2Ri h 0.3974x 2.984 × 10-3 x2 (7.1) Maximum width of the plot 2w close to the parabolic distribution of the roll bite length , given w q1 - 2 (w1-w2) x l r (w1 - w2) x2 L r 2 w - w1 - 2 (29.73 - 32.67) x 63.56 y 2 9.73 - 0.. 0925x - 7.297 × 10'4 x2 (7.2) Equation 7.2 is formed in such a way that it satisfies the boundary conditions w - w1at x 0 and w2 on x th and dw dx conditions, taken by default when analyzing or formulating the sheet rolling process. The projected area of the work piece-roll of the contact surface on the x-z plane is close to the semi elliptical form of width 2b at exit and 2b at any site at a distance x from the entrance, expressed as b2 2x L r - x2 L r Department of Mechanical Engineering, RCOEM, Nagpur 34 44. b 26.72 2 2x 63.56 - x2 63.562 (7.3) 7.2 Calculation of effective roll radius Effective roll radius Effective roll radius is derived from the maximum roll radius, R o , G roll clearance and maximum output width of 2w2 and exit area A2 as, R e r o 0.5 A2 2w2 - G (7.4) R e 180.25 - 0.5 1804.88 2 ∗ 17.67 - 11.5 R e 164.125mm The aforementioned equation 9.4 only works then only works then works only then when 11.5 R e 164.125mm the aforementioned 9.4 equation only works when the output is cross-sectional form rectangular. Equation 9.4 is based on the equivalent method of rectangular approximation, which transmits a non-rectangular cross-section to a rectangular width equal to the maximum cross-section width, while the pure cross sectional area is maintained equally. 7.3 Calculation and voltage speed For homogeneous deformation of the working part in the direction of height and width, the corresponding deformation component can be found as follows. The load deviation curve for the duct material is shown in the next digit. Hot hire includes a plastic deformation deviation of the working part - changing the length of the original length Figure 9.1: Attitude for Steel Department of Mechanical Engineering, RCOEM, Nagpur 35 45. d'dx x - final length - original length of the original length Integration, with a limitation from Li toL o ε - d'dx x L0 L I , th ln x - ln l l o (7.5) Note: '- ve' sign can be neglected for calculations because it indicates whether the voltage is due to voltage or compression. Thus, along the height of the work piece, ε y ln h1 ε y y y y 29.73 17.67 y 0.52 Similarly, the voltage in the direction of width, ε z y ln w1 w2 ε z 17.6 29.73 - 0.0943 The volume of the working part remains constant during the full rolling process. Thus, ε x x ε y ε z 0 ε x ε th ε z ε x (0.52 y 0.0943) - 0.4257 Neglect components of haircut deformation, effective voltage received as Department of Mechanical Engineering, RCOEM, Nagpur 36 46. ε g 2 3 ε x 2 y 2 y 2 ε g 2 th ε z 2 (7.6) ε g 0.5541 7.4 Calculating effective strain rate or effective voltage rate is voltage change rate, and given ε̄ ε t p (7.7) Where the ε is deformation and t p is the time it takes to paint or work a piece to go through the length of the contact. t p can be calculated in relation to t p 60× L r 2'N R eff (7.8) t p 60 ∗ 63.56 2' ∗ 79 ∗ 167.53'180.25 2 tp 0.047 Thus, voltage speeds can be calculated using an equation (7.8) as, ε̇ y ε t p 0.52 0.047 - 11,108 s-1 ε̇z - ε z t p - 0.0943 0.047 ε̇z , by the law of volume preservation, ε ẋ - ε ẏ and ε (7.9) ε ẋ 0.094 s-1 Result strain speed, Department of Mechanical Engineering, RCOEM, Nagpur 37 47. ε ṙ - ε ẋ 2 - ε ẏ 2 - ε No. 2 - 11.84 s-1 7.5 Calculating material properties in the mobile state of the above analysis and data, the working unit temperature on the stand is about 1000O C. Strain in the working part by height (off-direction) Work piece along height (off-direction) 11,108 To find the flow of voltage material in these conditions of deformation, temperature voltage rate, different equations can be used. The flow of stress through Shida's equations in these conditions can be detected. Since the installation of a rolling plant can be used to roll different types of steel, calculating the load flow for steel containing a different proportion of carbon and other constituents (properties mostly depend on the carbon content in steel). . 7.6 Shida's equation for finding flow stresses At a temperature of 1000O C and carbon content of 0.2%, σ p = σf ∗ fr ∗ f kgf mm2 (7.10) T = 1000 + 273 1000 = 1.273 Tp = 0.95 × C+0.41 C+0.32 = 1.1144

(7.11) For T ≥ Tp, σt = 0.28 ∗ exp 5 T − 0.01 C+0.05 = 13.664 (7.12) m = (−0.019C + 0.126)T + (0.075C − 0.05) = 0.12 n = 0.41 − 0.07C = 0.396 f = 1.3 × (5ε) n − 1.5ε = 1.118 (7.13) fr = (ε̇ 10) m = 1.013 (7.14) Thus, σ p = 15.47 kgf mm2 = 151.76MPa For T < Tp, the equations to be followed are: Department Of Mechanical Engineering, RCOEM, Nagpur 38 48. No 0.28 ∗ q (c, t) exp c 0.32 0.19 (c - 0.41) - 0.01 with 0.05 q (c, (t) - 30 (from 0.9) (T - 0.95 ∗ from 0.49 s and 0.42) ε̇ 10 m. ε̇ 100 m 2.4 . (ε̇ 1000) m/15 m . (0.081C - 0.154) T - 0.019C .207 - 0.027 C'0.32 From the above values and graph we can conclude that the load of flow for different steels with different percentages of carbon is almost the same in rolling conditions. Thus, given the average stress flow value as 152Mpa, deviatoric stress components can be detected. 7.7 Calculating the components of deviator stress Deviatoric stress components are those that must be exceeded under certain conditions to change the cross-section of the working part. These components in 3 directions x, y and z can be found by the rule of flow mises q x 2 3 σ σ̇ r ε ẋ Figure 9.2: The ratio of temperature flow voltage for steels on, what with this voltage speed Division of Mechanical Engineering, RCOEM, Nagpur 39 49. x 2 3 × 152 11.84 ×9,049 77.83 Mpa y 2 3 σ σ̇ p ε ẏ According to an earlier assumption made ×, inter-standard tension is ignored by the × inter-standard tension. Thus, the rear and front tensions are taken to zero. 7.8 Roll load calculation, torque and power 7.8.1 Load Roll F No 2 ∫ σ b dx L x 0 F 2 95.07 63.56 0 × 26.72 2 2x 63.56 x2 63.562 dx F No 126,810 KN Negative Signs that indicate signs that indicate the load is compressive in nature. 7.8.2 Roll torque MH 4 ∫ σ y b L x 0 (L r x) dx r e (F1 - F2) M y 4 95.07 ∗ 27.62 2x 63.56 x 63,562 (63.562 6 x) 63.56 0 dx M 6841.6 Nm 7.8.3 P - M ∗ V2 R e P - 6841.6∗1.32 0.164 and 55.07 kW 0 (expected) Department of Mechanical Engineering, RCOEM , Nagpur 40 50. To reduce the area of the working part requires torque, load and calculated power. The torque and load are provided by 2 rollers, which are connected to 2 universal shafts of joints. Thus, the load applied by each roll, 0.5 (126.81) Nm-63.4KN AND, Torque, applied by each roll 0.5 (6841.6) Nm 3420.8 Nm 7.8.4 Power, Supplied by shaft P, 2'NT 60 P and 2∗79∗6841.6 60 and 56.6 kW Thus, power loss 56.6-55.07 and 1.53 kW Department of Mechanical Engineering, RCOEM, Nagpur 41 51. Chapter 08 VALIDATION 8.1 Reaction calculation when bearing, to find the lifespan of bearings, it is necessary to calculate the load on the bearings. This load is a reaction to bearings due to the rolling load, torque, shaft weight, wobblers and other components. Thus, modeling the shaft and wobblers as a beam with supports at two ends and finding a reaction on bearings that can be seen as loops. Reactions are as follows: R A - 1424N and R B - 1589.48N Max BM -569979.7 N-mm 8.2 Mine Diameter Calculation (as code as ASME) No. 16 ∗ 103 xD3 ∗ qlt;9'gt; (Kt T)2 (KB M)2 T 3420.8 NM KT 1.8 M 569.98 Nm Kb 2.6 th min 0.3 per ht, 0.18 s ut for SAE 4340, Min (0.3 ∗ 470 , 0.18 ∗ 689) 124.02 mpa D3 - 16 ∗ 103 π ∗ 124.02 × (1.8 ∗ 3420.8) 2 (569.98 ∗ 2.6)2 D3 and 260121.62 mm3 D and 63.84 mm (140 mm (minimum diameter of solid parts of the shaft) are therefore safe. Department of Mechanical Engineering, RCOEM, Nagpur 42 52. Modeling the working length of the shaft using MDSolids 3.5, Mechanical Engineering Division, RCOEM, Nagpur 43 53. 8.3 Calculating the voltage of the haircut in the floor shaft of × π × × the torque and the moment D max 200 mm D min . 0.84) × (1.8 ∗ 3420.8)2 (569.98 ∗ 2.6) 2 max and 6.83 MPa zlt; 124 MPa thus, the hollow part of the shaft is also safe. Department of Mechanical Engineering, RCOEM, Nagpur 44 54. Chapter 09 CAD MODEL OF SHAFT AND OTHER PARTS The Head Input Model CAD Assembly Shaft was developed using SolidWorks (version: 2013) and exported to . IGS format. The Whole Shaft build consists of the following components: 9.1 The main assembly of the spline shaft is provided for two purposes - one, to assist in Adjusting the length of the shaft according to different applications; two, they prevent the relative sliding of the two parts of the shaft. To connect the universal joints to the gearbox and roller, wobblers are provided on both sides of the main assembly. 9.2 Wobbler (Gear Box side) This wobbler has two types of pads, namely flat pads and radial pads in its inner periphery. The pads are made of fiber and serve to absorb the impacts created during the mine. Figure 9.1: (above) the assembly of the main shaft. No, no, no. Blown View Figure 9.2: Wobbler (Gear Box Side) Department of Mechanical Engineering, RCOEM, Nagpur 45 55. 9.3 Wobbler (Roller Side): This wobbler is similar in design to its gearbox counterpart, except that it has a locaying pin that fits into a similar groove in the roller. This ensures that the roller and the shaft are perfectly aligned. 9.4 Yoke Build with Universal Connections: This assembly consists of split-yo-assembling eye type, eye-type ig assembly, and spider-assembling consisting of bearings. Figure 9.3: Wobbler (Roller side) Figure 9.4: Eye Assembly Figure 9.5: Blown Up View of Universal Collaborative Assembly Department Of Mechanical Engineering, RCOEM, Nagpur 46 56. 9.5 Yoke Ping: Yogo Contact is the main focus of this project. It is installed inside the hole in the split eye-type yoke and extends to a slot in the eye plate. Its main function is to limit the angle made by a universal connection; thereby preventing damage to the joint. 9.6 Needle bearings: Needle bearings used in this grade model IS 4215 RN1 - NB506225, meaning that the case is 50mm of internal diameter, 62mm of outer diameter and 25mm thick. The needle itself is located in two rows, the specification is 6 mm in diameter and 17.5 mm in height. Needle bearings are in direct contact with the spider's hand (inner), as there is no internal race in the dwellings. The need for two rows of bearings instead of one long bearing is due to the fact that the probability of failure as a result of the crushing of the bearing in two private arrangements is less. 9.7 Roller: Rollers are the most important components in any rental mill. In this case, there are two videos that are installed at a short distance; while the blank runs between them. The rollers are made of cast steel and have grooves on their surfaces. The curvature and depth of these grooves depend on the application. The outer diameter of the roller is 360.5 mm, and its surface has 5 grooves. One side of the roller is trained to fit in similar geometry in the wobbler. Figure 9.6: 9.7 pin drawing: Needle bearing with mechanical engineering department, RCOEM, Nagpur 47 57. 9.8 Spider Assembly: This assembly consists of a spider that has four hands on which four The bearings are installed. Two of these enclosures fit into the separated type of eye, and the other two fit into a type of eye plate that is mutually perpendicular to the split eye-type yoke. Figure 9.8: Video. Figure 9.9: Spider Build Figure 9.10: Full Shaft Assembly Department Of Mechanical Engineering, RCOEM, Nagpur 48 58. The material parts of the Assembly reviewed in the Finite Element assembly Below: Table 9.1: Materials from various components of The Material Young's Modulus (MPa) Poisson Coefficient Density (kg/m3) Shaft Steel 2.1 e 5 0.3 7850 Eye Plate Steel 2.1 e 5 0.3 7850 Wobbler Steel 2.1 e 5 0.3 7850 Yoke Pin Steel 2.1 e 5 0.3 7850 Mechanical Engineering Department, RCOEM, Nagpur 49 59. Chapter 10 MESH GENERATION 10.1 Geometry Cleaning: The imported CAD model is cleaned for its geometry, thus forming a good input for modeling the final elements. Initially, surfaces are developed from hard components, after which solids are removed as the end elements are generated from surfaces. Global tolerance to cleanup is set as 0.01. The model is checked for free edges and is cleaned with switching, replacement and equivalence operations with appropriate cleaning clearances. The model is then tested for the absence of surfaces that can be identified at free edges, and new surfaces are created in these locations. The model is then checked for non-multiple edges, where there are more than two surfaces separating the edge, which may indicate the wrong connection. Duplicate surfaces are also checked with clearance tolerance of 0.01 and then removed. The fillets of minor importance are removed, thus simplifying the geometry. The surfaces of bolts, nuts and washers are removed on the model of the appropriate types of elements in the process of modeling the final elements. Fillet of small parts removed Figure 10.1: Cleaning geometry - Removing fillet of small parts Toggled Edge Figure 10.2: Geometry Cleaning - Toggled Edge Division of Mechanical Engineering, RCOEM, Nagpur 50 60. 10.2 Component grid in HyperMesh 12.0: After cleaning the geometry, all surfaces are mixed with medium-sized Tria and R-Tria elements to form a closed grid. 3D Tetrahedral Elements are created from a closed 2D grid to form solid components. Property collectors are created for each component with a PSOLID map and are assigned accordingly. Figure 10.3: Net model Shaft Assembly Figure 10.4: Net model eye and slab assembly Figure 10.5: Net Model Wobbler Build Figure 10.6: Mesh Model Eye Plate and Yoke Pin Department of Mechanical Engineering, RCOEM, Nagpur 51 61. 10.3 Item quality check: The geometric quality of all 1D, 2D, 3D elements of the model is checked. The model is also checked for proper connectivity and duplicate elements. Table 10.1: Item quality 1 D Elements Free 1 D Elements Rigid Loop Double Dependence 2 D Elements Warpage 15.00 Aspect 10.00 Length 2.00 Jacobian 0.5 Tria Minimum Corner 20 Tria Maximum Angle 20 120 3 D Elements Tet Collapse 0.2 Length 2.00 Tria Minimum Corner 20 Tria Maximum Corner 120 Figure 10.7: Grid Model Valt Assembly Department of Mechanical Engineering, RCOEM, Nagpur 52 62. 10.4 Border conditions: DOFs wobblers are limited to fixing the face of wobblers on both sides. The limitations are shown by blue triangles (figure) 10.5 Load steps: The mesh assembly is analyzed for the following: 1) SOL 103: Limited modal analysis - to determine natural build frequencies and mode forms. - The EIGRL map for the first 4 mass normalized modes is indicated to perform a cost analysis of Real Eigen using the LAN-OS 2) SOL 101: Static Analysis - by applying the moment to the shaft, identified areas of higher voltages. Figure 12.9: 3D Element Representation Figure 12.10: Traditional Representation Element of the Department of Mechanical Engineering, RCOEM, Nagpur 53 63. Chapter 11 MODAL ANALYSIS 11.1 Introduction: Modal Analysis is the process of identifying the system's inherent dynamic characteristics in forms of natural frequencies, damping and modenic form, and using them to formulate a mathematical model for its dynamic behavior. The dynamics of the structure are physically decomposed by frequency and position. It is based on the fact that the vibrational reaction of the linear dynamic system of time can be expressed as a linear combination of a set of simple harmonic movements called natural vibration modes. 11.2 Mode Forms: The formulated mathematical model in the process of modal analysis is called a modal model of the system. Natural vibration modes are inherent in a dynamic system and are fully determined by its physical properties (mass, rigidity, damping) and spatial distributions. Each mode is described in terms of its modal parameters: natural frequency, modal damping factor and characteristic pattern of bias, namely the shape of the regime. The mode form can be real or complex. Each corresponds to the natural frequency. The degree of participation of each natural mode in the overall vibration is determined by both the properties of the source of arousal and the forms of the system mode. Mode 4 Mode 1 Mode 2 Mode 5 Mode 3 Mode 6 Figure 11.1: Form Mode Division Of Mechanical Engineering, RCOEM, Nagpur 54 64. 11.3 One degree of freedom: a system with one degree of freedom (SDOF) is described by the following equation: mẍ (t) - cẋ (t) qx (t) - f't), where m is mass, 'c'is the damping factor, and 'k'is - rigidity. This equation states that the sum of all forces operating on the mass of 'm' should be zero with f(t) is an externally used force, inertial force, -cẋ (t) is a (viscous) damping force and -kx (t) is a regenerating force. Variable x(t) means the position of the mass m relative to its equilibrium point, i.e. the position of the mass m when f(t) 0. Transmission function H (s) between movement and force, X (s) - H (s)F (s) , given, H (s) - 1 ms2 - cs k Roots denominator transmission function, d (s) - ms2 and cs'k, are the poles of the system. In mechanical structures, the c damping ratio is usually very small, which results in a complex pair of poles, No σ ± i'm Unsusilized natural frequency is given fn/2 π where n q k/m When the mass m is added to the original mass of the m structure, its natural frequency decreases to, Although very few practical structures can actually be modeled by the SDOF system, the properties of such a system are important because the properties of the more complex MDOF system can always be presented as a linear 11.2: SDOF System Engineering Department, RCOEM, Nagpur 5 Superposition of a number of SDOF characteristics (when linear time system- invariant). 11.4 Multi-stage degree of freedom: Multiple Freedom Systems (MDOF) are described by the following equation: Mẍ (t) - Cẋ (t) H (s) - Ms2 Cs CK1 - N (s) d (s) Where , N (s) - adj (Ms2 - Cs q K) And, d (s) - det (Ms2 - Cs and K) When damping is small, the roots of the characteristic polynomial d(s) are complex pairs of conjuged poles∗, N m, with nmthe number of system modes. The transmission function can be rewritten in the form of pole residues, which is considered a modal model. H (s) - ∑ R m M M-1 and R M∗ s-m∗ Remnants of the Rm Matrix, m No. 1...... Rm, defined, R m lim s→ m (s) yu m) Can show that the R incorrectly rank one matrix means that R Mcan be decomposed as, Figure 11.3: MDOF System Department of Mechanical Engineering, RCOEM, Nagpur 56 66. with ψm vector, representing the form of mode mode m. Complete matrix of transmission function is completely characterized by modal parameters, i.e. poles, mm and ± i'm, and vectors of the form of ψm, m and 1...... Nm. 11.5 Free - Free analysis: Free analysis is performed to extract rigid body modes from a simulated assembly. All restrictions specified in the model are removed before performing a free - free analysis. This analysis is also used to verify that all components are connected to each other. Natural frequencies the first six fundamental regimes were about zero. Figure 11.4: Mode 1 (above): Translation of X-Axis Mode 2 (below): Translation on the Y-axis of the Department of Mechanical Engineering, RCOEM, Nagpur 57 67. Figure 11.6: Mode 4- Rotation on X-Axis Figure 11.7: Mode 5- Rotation of Y-Axis Figure 11.5: Mode 3- Translation of the Axis Department of Mechanical Engineering, RCOEM, Nagpur 58 68. Figure 11.8: Mode 6- Rotation of q- axis 11.6 Normal mode analysis: With limitations in the places bearing the eye plate, the natural frequencies and shapes of the assembly mode are as follows: Figure 11.9: (above) Normal Mode 1, (below) Normal Mode 2 Department of Mechanical Engineering, REMCO, Nagpur 59 69. Pairing the surface of the pin with a universal joint eye is observed to be very tense due to the upper-down movement and therefore has a greater chance of failure in the passage of several work cycles. Figure 11.10: (above) Normal Mode 3, (below) Normal Mode 4 Figure 11.11: Tense Areas of the Eye Plate Division of Mechanical Engineering, RCOEM, Nagpur 60 70. 11.7 Calculating the natural frequency First finding the equivalent diameter of the shaft, as it is divided into parts of different diameter and length (e.g. flanks, splicing parts, wobblers, etc.) Because the same torque acts on each part of the shaft, the same turn is generated. TL GJ - ∑ TL i GJ i i i (17.1) d - 191.99 mm, 0.192 m I - π 64 d4 (17.2) I π 64 (0.195)4 and 6.67 × 10-5 Since chiller fixed, miller is miler, 4 it has no deviation, hence we only consider deviation due to self-weight in weight. The maximum deviation from the shaft received is 0.01484 mm fn - 1 2 g δ - 0.4985 √δ (17.3) fn - 129.4 Hz (17.4) Figure 11.12: shaft loading, with the fixation of wobblers Figure 11.13: The deviation of the shaft due to its independent weight. Department of Mechanical Engineering, RCOEM, Nagpur 61 71. Mode No. 1 2 3 4 Natural frequency 121.9 122.3 305.5 313.9 Natural frequency by analytical calculations is 129.4 Hz. The margin of error compared to the natural frequency of Mode 2 is 5.8%. 11.8 Static stress analysis with limitations in place, input point value 6400 Nm applies to the main shaft of direction X and areas of high stress locations (Von-Misans stress) are determined. It has been established that the load on the igo contact is 15 MPa on average. Compared to its yield of 210MPa, it is much smaller. Thus, the possible conclusion is that the needle pin gets failed due to wear and tear. Figure 11.14: Stressed areas of pins and eye plate Table 11.1: Natural Frequency (by Hyper Mesh) Department of Mechanical Engineering, RCOEM, Nagpur 62 72. Chapter 12 WARNING AND WARNING 12.1 Conclusion - From the above analysis, the conclusion is that the igo pin has very less stress acting on Tensions have a magnitude of 15MPa on average. Contact material SAE 1045. Tensions for the contact material are tolerable and the pin is safe in the crush, tension, haircut and bend. Mainly because of wear and tear, Yoke Pin is unable to be used. The hole for the pin has voltages equal to 2MPa. The material is strong enough to withstand these stresses. - Both the pin and the hole for the eye plate wear out because of the jerks of the shaft. The ducts into the shaft are mainly caused by the impact of the blank, the compression of air gaps in the blank, uneven cross sections, surface irregularities on the blank, etc. - the pin and the hole in the eye plate need some treatment for durability. The natural frequency of the shaft is 122 Hz. The error compared to the software results is 5.8%. 12.2 Modifications to reduce component wear: - Use of a bush between a pin and a hole that wears out and therefore will retain the wear-hazardous components of the assembly. o Using the bush will reduce repair time. Just a bush has to be replaced. Use a press fit coating on the pin and hole. This will prove to be a very effective time. o Simply changing the coating on the pin and hole will reduce repair time and increase the time between repairs. o The coating material can be appropriately selected depending on the position of the stand, the frequency of failures, etc. - A review of the welding materials and the process currently used to repair the damaged parts. o Using different materials to fill the metal in the gap that forms in the hole of the eye plate. o The time it takes to repair may not change, but the time between failures can certainly increase. Department of Mechanical Engineering, RCOEM, Nagpur 63 73. Chapter 13 FUTURE SCOPE Project can be expanded in different horizons. The changes proposed in section 12.2 need to be carefully analysed. Changes need to be analyzed in terms of cost, efficiency of time, adaptation of workers, etc. Further dimensions for the project may be: - The design of the bush between the pin and the hole that wears out and therefore maintain the wear of the prone assembly components. Factors to consider for this: o Design changes. o Material bush. Cost analysis. The design of the press fits the coating on the pin and hole. Factors to consider: o Design modifications. o Stress analysis of the new design. The tolerances that will be given to the components will be a serious problem. Review of welding materials and the process currently used to repair damaged parts. o Various welding materials should be used on the pin and hole to increase the time between the two failures. Reducing bumps and jerks, other on the shaft. Ways to reduce shock need to be devised. o At least shock for some different components that is strong enough to take them to be developed. Department of Mechanical Engineering, RCOEM, Nagpur 64 74. (APPENDIXS USED) SOLIDWORKS 2013 (Modelling CAD) SolidWorks is a solid Para-based fashion designer and uses a feature-based para-based para approach to create models and assemblies. The options are the limitations of which determine the shape or geometry of a model or build. Options can be both numerical parameters, such as line length or circle diameter, and geometric parameters such as tangent, parallel, concentric, horizontal or vertical, etc. numerical parameters can be linked to each other through relationships that allow them to capture design intentions. Creating a model in SolidWorks usually starts with a 2D sketch (although 3D sketches are available to power users). The sketch consists of geometry such as dots, lines, arcs, conics (except hyperbole), and splines. Dimensions are added to the sketch to determine the size and location of the geometry. Relationships are used to define attributes such as tangent, concurrency, perpendicularity and concentricity. The parametric nature of SolidWorks means that sizes and relationships control geometry, not the other way around. Dimensions on the sketch can be controlled independently, or by relationships with other parameters inside or outside the sketch. In the assembly analogue of sketchy relationships are comrades. Just as sketch relationships define conditions such as tangent, concurrency, and concentricity in sketch geometry, build mates define equivalent relationships in relation to individual parts or components, making assemblages easy to construct. SolidWorks also includes additional advanced pairing features such as gear and cam follower mates, which allow simulated assembly gears to accurately reproduce the rotational motion of the actual train gear. Finally, the drawings can be created either from parts or from assemblies. Views are automatically generated from a solid model, and notes, sizes, and tolerances can be easily added to the drawing as needed. The drawing module includes most paper sizes and standards (ANSI, ISO, DIN, GOST, JIS, BSI and SAC). Department of Mechanical Engineering, RCOEM, Nagpur 65 75. HYPERMESH 12.0 (For Mesh) Altair HyperMesh is a high performance end element preprocessor to prepare even the largest models ranging from import geometry CAD to export analysis run for various disciplines. HyperMesh allows engineers to receive high-quality grids with maximum accuracy as soon as possible. A complete set of geometry editing tools helps effectively prepare CAD models for the grid process. Grid algorithms for shell and the elements provide a full level of control, or can be used automatically. Altair Altair the technology of mesh hundreds of files is exactly in the background to meet user standards. HyperMesh offers the greatest variety of solid grid capabilities on the market, including domain-specific techniques such as SPH, NVH or CFD grids. The long list of CAD formats provides a high level of CAD compatibility. Altair connector technology automatically collects individual parts with their Finite Element view. HyperMesh is fully configured. An extensive API library can be used to automate repetitive tasks or for complex mathematical operations to generate models. With a focus on engineering performance, HyperMesh is the preferred user medium for: - Solid Geometry Modeling - Shell Grid - Model Morphing - Detailed Model Setting - Surface Geometry Simulation - Generation Solid Grid - Automatic Generation Medium Surface - Package MDSolids (SFD, BMD Solver) MDSolids is a multifaceted software that offers The MDSolids graphical user interface makes the software visually appealing, easy to use and very intuitive, so that students can focus on problem-solving concepts rather than on the work of software. Department of Mechanical Engineering, RCOEM, Nagpur 66 76. NASTRAN (Solver) NASTRAN is the ultimate element of the analysis (FEA) program that was originally developed for NASA in the late 1960s with funding from the United States government for the aerospace industry. Mac Neal-Schwendler Corporation (MSC) was one of the main and original developers of the public domain NASTRAN code. The NASTRAN source code is integrated into a number of different software packages that are distributed by a number of companies. MSC Nastran is an HPC-enabled structural analysis application used by engineers to perform static, dynamic and thermal analysis in linear and non-linear areas, complete with automated optimization and award-winning built-in fatigue analysis technologies. From high-performance computing capabilities to the high degree of certainty it provides, MSC Nastran is designed to give increased awareness of component behavior. Accurately and quickly predict the complex behavior of the product. Find conflict design at the beginning of the design cycle. Reduce the number of design changes. - Make compromise studies of performance and reliability. Opportunities: Predict the life of the product, optimize the design. Advanced non-linear multi-physics simulation Dynamic response to MSC integrated structural solutions for linear and linear and Computing makes it easier to reuse models, which saves a lot of time in pre-processing and allows us to standardize data sharing for body models in collaboration with other departments or external suppliers, Department of Mechanical Engineering, RCOEM, Nagpur 67 77. APP (ABOUT SUNFLAG IRON AND STEEL LTD.) Sunflag Iron and Steel Co. Ltd. is a prestigious division of SUN FLAG GROUP. It has established a state-of-the-art integrated plant in Bhandara, India. The plant has the capacity to produce 360,000 tons of high-quality special steel using liquid cast iron and spongy iron as the main input. Sunflag Steel meets the needs of various industries such as cars, railways, defense, agriculture and engineering, and is approved by all international national OEM. Specialty steel is produced in accordance with Indian and international standards confirming DIN/SAE/AISI/EN, JIS, GOST, IS and Honda Motors patented varieties. The plant includes 2.62,000 tons per year of direct plant reduction, for the production of spongy iron for consumption in captivity in a steel smelter. This store includes a 50/60 ton ultra-high power electric arc furnace with an eccentric bottom arrangement; The bucket of the refinery, the tank degassing and the Bloom/Billet dual casting radius with AMLC and EMS and T type tundish. The blanks, made at the steelworks, are rolled at the state-of-the-art 20 stands mannesmann Demag Designed. This mill has a dual type of fuel walking fire-heating furnace, rapid change of roll objects, 65 meters walk and waiting type of modern cooling bed and above all computerized control of the process of linking and managing different stages. In its short period of creation in 1989, SUNFLAG STEEL has established itself as a major global force. This modern complex, pulsating with world-class technologies, expert human resources and a desire for excellence, has created a clear niche in alloy steel, stainless steel and micro-alloy steel and has reached the position of market leader in the segment. Today SUNFLAG STEEL has also started exporting and regularly receives prestigious orders from Japan and many other countries in the Far East, Afro-Asian and Middle East. The 30 MW captive power plant has already been put into operation using gases. Department of Mechanical Engineering, RCOEM, Nagpur 68 78. The Bar and Mill Section (BSM): 60 tons per hour, 20 stand, 20 tall continuous Mannesmann designed rolling plant equipped with duel fuel walking hearth-like re-heating furnace, management process through ABB automation, online evaluation system, closed chain television system for harvesting and bar movements. Variable Mill Reduction (VRM) is installed after booth number 20 for the production of a product with close tolerance. Product sizes - round (diameter 15-53 mm), Hex (13 to 38.5 mm), WRD 12 to 38 mm), WRB (4.6 to 16 mm) Mechanical Engineering Division, RCOEM, Nagpur 69 79. HELP Patents 1. Edmund. Morewood, Improvement in in Iron and Copper, U.S. Patent 3746, September 17, 1844. Books 2. Kalpakjian, Schmid, Manufacturing Processes for Engineering Materials, 5th. 2008, Pearson Education ISBN No 0-13-227271-7. 3. George E. Dieter Jr. Mechanical Metallurgy. MCGRAW-HILL BEECH COMPANY 4. Youngseog Lee, Rod and Bar Rolling, Theory and App, Mercel Decker Inc. 5. George T. Halmos, Roll Formation Director, CRC Press, Taylor and Francis Group, 2006. 6. Lenard J.G., Piterzyk M., Cser, L, Mathematical and Physical Product Properties Modeling, Elsevier Science Ltd., 1999.Pp. 74-76 Documents 7. Paluh J H, Hills G R, Wojtkowski T C. Designing and selecting universal shafts for rolling plants 8. 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