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EFFECT OF SPEED AND SKEWING ANGLE IN CONTROLLING AXIAL THRUST OF ROTARY KILN

A THESIS IN PARTIAL FULFILLMENT OF THE REQUIREMENTS FOR THE DEGREE OF MASTER OF SCIENCE IN MECHANICAL ENGINEERING

(MECHANICAL DESIGN STREAM)

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ABSTRACT

Usually the axial thrust of a rotary kiln has been controlled by a hydraulic thrust device that constantly pushes the kiln up against gravity. However, the life of the device will be deteriorating earlier expected if other mechanisms are ignored to consider incorporating so that magnitude of the thrust is shared with because the thrust load varies for different factors to extreme values that could damage different components of rotary kiln system including the hydraulic thrust. Using static and dynamic modeling together with ANSYS and CATIA software it is possible to show the effects of speed and skewing angle in controlling the axial thrust of rotary kiln. And this will help for the development of technology which will further be assisted by automation by considering the different varying factors of operating conditions that include mainly the surface condition of support rollers, slope of the kiln, rotational speed of the kiln, capacity of the rotary kiln, stability of the support roller alignment, roller lubrication, proper functioning of the hydraulic thrust device and others. After going through all these variables it is easy to come up with a particular setup that guides the operator/maintenance crew to maintain the specific condition to help moderate the axial thrust of the rotary kiln. Because of time and resource limitation this thesis is mainly concerned with the effect of rotational speed and skewing angle of a kiln which are the major factors affecting axial thrust and it will use analysis method of mathematical and ANSYS modeling. And with this we can prolong the lives of different components, increase production uptime, and reduce power consumption

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CHAPTER ONE

1. INTRODUCTION

The rotating kiln shell on a floating tire rings is supported by paired rollers as many as one pair for every 20m length of kiln shell which must have the same inclination as the kiln shell so that the support rollers and kiln tire rings make a line contact. The contact pressure with the rotational motion tends to create a huge axial thrust load that can have a direct impact on the temperature rise of support roller bearings and also causes other mechanical component failures. The major working conditions (or simply the variables) that result in complex phenomena in controlling the axial thrust of the kiln include capacity of the rotary kiln, rotational speed of kiln, conditions of contact surface between support rollers and tire rings, inclination of all rotating and static components of kiln system, stability of the support roller alignment, roller-tire contact surface lubrication, proper functioning of the hydraulic thrust device and others. Identifying the relationship among each variable will be one of the steps in devising mechanisms ultimately to control axial thrust of the kiln. Presently rotary kilns particularly in our country which have axial lengths up to 66m are given nearly from 3 to 3.5% inclination with the combination of rotational motion of kiln to allow material flowing out in one direction. The rotary kiln consists of a tube made from steel plate, and lined with firebrick. The tube slowly rotates on its axis at between 30 and 250 revolutions per hour. Efficiency of kiln operation is usually evaluated by running the kiln for as much long period of time as possible. That is why rotary kilns run 24 hours a day, and are typically stopped only for a few days once or twice a year for essential maintenance. This is because heating up and cooling down are long, wasteful and damaging processes. For example, the starting cost of a 3000tones/day capacity kiln operation could roughly mount to ETB 1.5million as of now. That is why uninterrupted runs as long as 18 months have been achieved and highly appreciated in some of the kiln operations around the world. However, there are also on the contrary, side effects particularly on the maintenance department when the kiln runs for considerably long period of time. Distortion which is a bent axis creates a multitude of problems depending on their magnitude and location. This problem arises usually due to a high temperature exposure of the kiln for an extended period. Premature refractory failure due to mechanical instability often results. Since the shell is no longer rolling as a straight cylinder it is subject to additional bending stresses as its weight cycles from one roller to another. In extreme cases, a portion of the load shifts from pier to pier. The rollers, shafts, bearings and bases, which are designed to support a portion of the total rotating load, now bear a reduced load for part of each revolution and a higher load for the balance of each cycle. A badly bent shell can induce extremely high load peaks during each rotation cycle. In addition to the shell problems, this overloading can lead to shaft failure, hot bearings or even support base damage. The axial thrust of the kiln which is the core issue of this thesis will also be intensified on support rollers and hydraulic thrust device. With all these problems at hand if there are no designed courses of action that are to be taken based on the co-relations of every variable, a catastrophic failure will be expected at any time. That is why this thesis focuses on determining the effects of rotational speed of a kiln and skewing angle of support rollers so that the result will help balance and optimize the axial thrust of the rotary kiln.

1.1. BACKGROUND

So far much has been contributed for the immense development of rotary kiln technology but less for axial thrust in particular

- Linear laser scanning of kiln shell (2000)
- Statistical methods of circular deviations elimination from calculation of kiln shell radial run out(2001)
- Analysis of cyclical change of support loads through measurement of roller shaft deflection(2005)
- Analysis of kiln shell temperature profile influence on kiln crank(2005)
- The method of 3D visualization of kiln shell deformation(2008)
- Full numerical kiln modeling system which enable to calculate and visualize kiln mechanical key parameters(2010)

It is the axial resistance exerted by one or more components against gravity that is resulted from inclination of the rotary kiln and as a result the component in charge will be mechanically affected due to the extreme pressure.

Factors affecting axial thrust

1. Skewing of paired support rollers
2. Rotational speed of the kiln
3. Surface conditions of support rollers and tires
4. Controlling slope changes of all rotating parts
5. Operational load of the kiln

1.2. Purpose

This thesis is mainly concerned with determining the effects of rotational speed and skewing angle of support rollers of a kiln which are supposed to be the major factors affecting axial thrust. The speed and skewing angle variation of the kiln rotation is assumed to have a significant alternating axial pressure on different components of rotary kiln. For example kiln thrust rollers, inlet and outlet sealing, slide bearings of kiln support rollers particularly at contact points of bearing collar and shaft shoulder, surfaces of tire ring and support rollers are all affected by the axial thrust of altering kiln rotating speed and skewing angle. Knowing and understanding the effects of speed and skewing angle will help to device a mechanism to control the axial thrust. And with this we can prolong the lives of different components, increase production uptime, and reduce power consumption

1.3. OBJECTIVE

General objective

- Objective of the thesis is determining the effects of rotational speed and skewing angle of support rollers in controlling axial thrust of a cement rotary kiln

Specific objectives

- Determining the effect of rotational speed
- Identifying which pair of support rollers makes significant change to control thrust when they are subjected to skewing

- Determining the effect of roller skewing quantitatively
- Comparing effects of the two factors

1.4. STATEMENT OF THE PROBLEM

Determining the effects of rotational speed and skewing angle on axial thrust of rotary kiln using static and dynamics modeling based on operating and design conditions of rotary kiln so that optimal axial thrust on support rollers and thrust device can be obtained whenever adjustments are made.

In the earlier times, axial thrust was controlled by skewing of the support rollers. However, nowadays a hydraulic thrust roller has been in use to support the axial thrust. Still the problem is not completely solved for the fact that the components of hydraulic thrust system usually fail due to extreme pressure of the rotary kiln especially during full load operation

1.5. ORGANIZATION OF THE PAPER

The rest of this thesis is organized as follows. Chapter 2 gives a description of literature review which delivers information about an overview of what has been said, who the key writers are, what are the prevailing theories and hypotheses that have finally been converted to technology, what questions are being asked, and what methods are appropriate and useful relevant to the topic of the thesis. The third chapter describes the materials, methods and conditions of the thesis. Chapter 4 is all about reporting the findings. It has summarized results obtained and discussed figures and tables of the result. The last chapter or the fifth one has included conclusion, possible recommendations and future works of the writer. Finally references and appendix have been included.

CHAPTER TWO

2. LITERATURE REVIEW

2.1 Introduction

[5]Even though a rotary kiln could be categorized under rotor dynamic equipment a particular attention should be given to, for the fact that the physical nature appears to be huge and heavy. The mass of a typical 6 x 60 m kiln, including refractory and feed, is around 1100 tons. The effect of a slight misalignment could cause a catastrophic failure. The contact pressure between the bearing collar and shaft shoulder of a support roller becomes so high that bearing temperature will immediately rise to an unfavorable condition. This is due to the axial movement of the rotary kiln usually against gravity. Cause of the axial movement depends on one or more of a number of variables. Usually actions are taken independently not in an integrated way without taking the co-relation between variables into consideration. This would take days, weeks, even months before getting the real solution. This is because the different actions taken are trial and error. As of today there have not been devices formulated to easily apply in getting into a course of action that enables us to control the situation so that the production interruption can be minimized to a great extent.

[1]The roots of Geoservex date back to 1978, when the company founder, MSc Eng. Bolesław Krystowczyk, was given the challenge to measure rotary kiln alignment. Firstly, he applied a classic attitude and took the kiln measurements during machine stoppage. He quickly noticed that as soon as the kiln was started up its cold state adjustment did not give satisfactory effects. The support system geometry appeared to behave differently when compared to its hot condition. The findings induced him to do further researches and studies for the reasons of the described effect. This led to the conclusion that measurements concerning such a type of machine should be performed during the kiln operation. Then, he undertook a challenge to work out a proper rotary kiln alignment measurement and support rollers' location method while the machine was operating. Research and design works lasted two years. In effect, as the first in the World he worked out the method that was later called "Hot Kiln Alignment". Nowadays, nobody undermines the principle of such an attitude but then it was a true technology breakthrough. In the 1980s, the method was implemented and applied to dozens of kilns. Bolesław Krystowczyk published principles listing the advantages of his solution in periodical “Zement Kalk Gips” (1983). The method was widely discussed and adduced in “Rock Produckts” (1987 & 1989) and “Wermessungs wesen und Raumordnung” (1986) magazines. An innovative character of the solution won for him an international patent and the author himself wrote a doctoral thesis in 1980 to be granted the title of the Doctor of Technical Sciences.

At that time, the problems with misalignment of multi-support kilns were so common. Despite of fact that Poland was a socialist country, alignment services based on PhD Eng. Bolesław Krystowczyk’s method were ordered by cement plants from such countries as the USA and Canada. True business breakthrough took place in 1990, right after a system revolution in Poland. Then, private company Geoservex was established and its founder gained an access to global market. At the same time, the method of kiln alignment at operating conditions was quickly gaining supporters and followers including leading kiln producers. Based on the same principle of measurement in operating state new solutions were worked out. In the next years, obvious new needs and new technical possibilities of development of kiln parameters diagnostics were appearing. In 2000, the founder’s son, Zbigniew Krystowczyk, joined the team of engineers. Combining staff experience with his own knowledge about world technical news started to introduce solutions such as linear laser scanning of kiln shell (2000), statistical methods of circular deviations elimination from calculation of kiln shell radial run-out (2001), analysis of cyclical change of support loads through the measurement of rollers’ shaft deflection (2005) and analysis of kiln shell temperature profile influence on kiln crank (2005). In 2007 Zbigniew Krystowczyk took up the position of Managing Director and the company’s main strategist. He noticed the potential of numerical modeling and the possibility of its application in the diagnostics of rotary kilns and appointed a new research team to work out the method of three-dimensional visualization of kiln shell deformation (2008). After that the team worked out full numerical kiln modeling system which enables to calculate and visualize all kiln mechanical key parameters (2010). The system bases on the Finite Element Method. Geoservex will continue its service development hoping that the offered scope of service meets customers' needs as properly fitted roots of girth gear and pinion.

Kiln as a system generates an axial forces depending on the natural gravitational movement down that is balanced by axial forces coming from rollers' skewing, inclination and surface shape. Thus, mechanical kiln balance depends on many factors. During kiln inspection all the influences are analyzed at the same time and the whole system is adjusted by paying special attention to all factors optimization. Theoretically, 30 % of the axial force should be balanced by the skewing of support rollers and their axial forces. If hydraulic system exists on the kiln, it should carry 70 % of the axial force and its pressure should be in the range of 40 – 60 Bar. Nevertheless, the adjustment process requires an individual examination of each case and that is why the task must be carried out by a specialized and experienced expert.

The axial thrust of kiln affects the following components of the system

- Both inlet and outlet sealing
- Surfaces of tire rings and support rollers
- Slide bearings of support rollers
- Gear and pinion contact surfaces
- Drive unit overloading
- Retaining rings

2.2. How Thrust Affects Each Component

2.2.1. Inlet sealing

[5]Inlet sealing plays an important role in protecting material fall out and most importantly preventing entry of false air that has a huge negative impact on productivity of rotary kiln. This is accomplished by adjusting predefined range of clearance between the stationary sealing and rotating kiln. However, because of continuous upward thrust, kiln inlet sealing is exposed to considerably fast wear. Since wear by nature means removal of material particles from the body of a component, a continuous removal of material will make the sealing lose its functionality. In addition to wear, the pressure exerted by upward thrust of kiln on the sealing will damage fixtures even structure of the system.

The following mechanical actions initiate upward movement of kiln

2.2.2.Outlet sealing

[5]Usually kiln tends to move axially downward due to gravity that results from the 3% inclination. It is easy to imagine how big this thrust load on the outlet sealing would be for the fact that the total mass of fully loaded rotary kiln climbs to 700ton.

The surface conditions of contacts between support rollers and tire rings have also a huge impact of the downward thrust on outlet sealing. When the roughness values of the surfaces decrease, it implies that there sounds to be more slippage between the contact surfaces. Since support rollers remain axially stationary this slippage means the axial movement of tire ring together with rotary kiln towards the outlet and produces a wearing and/or damaging thrust on the outlet sealing.

2.2.3.Surfaces of Tire Rings and Support Rollers

[4]When two bodies having curved surfaces are pressed together, point or line contact changes to area contact, and the stresses developed in the two bodies are three dimensional. Typical failures are seen as cracks, pits, or flaking in the surface material.

Wear conditions of surfaces of riding rings and support rollers:

- Convex and concave wear patterns
- Conical/tapered wear on riding ring/support roller diameters
- Uneven wear conditions
- Horizontal/diagonal marks on contact surfaces
- Surface spalling and high stress concentrations
- Broken edges on riding ring or support rollers
- Wear on riding ring thrust faces and thrust rollers

Convex/concave wear is the most common form of wear on the surfaces of riding rings and support rollers. The wear is a result of the support roller adjustment procedure employed to control the axial thrust of the kiln. The very act of adjusting the support rollers to control the axial thrust of the kiln will generate a convex/concave wear pattern between the riding rings and the support rollers.

The extent of the wear is directly proportional to the amount of support roller adjustment needed to control the axial thrust of the kiln. The greater the amount of bearing movements made to control axial thrust of the kiln will actually accelerate the wear on the surfaces of the riding rings and support rollers. As the contact face of the riding rings and support rollers decrease, the hertz pressures on these surfaces increase and subsequently, wear accelerates at a much higher rate. If the hertz pressures increase to a high enough level, surface spalling or metal flaking will occur. As the surface contact is lost, support roller adjustments also will increase to gain the same axial control Convex / concave wear and kiln operations:

- The axial thrust of the kiln will be unstable
- Individual support roller thrust can be excessive
- Increased wear will increase support roller adjustments
- Increased roller adjustments increase kiln drive amperages
- Increased drive amperage increases drive component wear
- Operating costs of the kiln increases

2.2.4.Support roller slope misalignment

In correct adjustment of the support rollers to control the axial thrust of the kiln(E.g. support rollers excessively skewed in opposite directions), conical wear, kiln alignment and support roller adjustment. The existence of conical wear patterns on the surfaces of riding rings and support rollers has an adverse effect of the alignment of the riding rings on individual piers for the following reasons.

The riding ring will tilt towards the smaller diameter and cause misalignment between the shell axis and riding ring axis. The kiln is designed so all components are rotating on the same axis. For example, if the kiln is set at an elevation slope of ½/ft, the rotational axis of the kiln shell, the riding ring and the support rollers need to be on this slope for ideal operating conditions to exist. Furthermore, the kiln shell axis and the support roller axis must be parallel in the horizontal plane for all the component parts to operate ideally. If for any reason, the surfaces of the riding ring or support rollers develop conical wear, the axis of the riding ring will not be aligned with the axis of the shell. The result will be a high thrust placed on the retaining blocks or rings that restrict the axial movement of the riding ring. In addition to this, a misalignment will occur between the riding ring ID and the filler bar OD exaggerating the wear between the frictional surfaces. Both of these conditions are continuous problems in the maintenance of the kiln. Excessive wear of retaining blocks/rings will cause the riding ring to move axially and ride off center on the support rollers. As the riding ring ID wears against the filler bars, a large gap will generate between the riding ring and the filler bars and result in higher kiln shell ovalities. In some cases, the conical wear condition on the surfaces of the riding rings and support rollers will cause problems with the axial thrust of the kiln and the individual thrust of the support rollers. If the surfaces of the riding rings and support rollers have excessive conical wear, the kiln may not be able to move freely in the axial directions as operational changes occur. This will cause higher thrust loading on individual support roller bearings and thrust rollers. It is not uncommon for the kiln drive amperage to run high in this situation and subsequently causes accelerated wear on the drive components. On some kilns, the signs of maintenance problems will occur if the riding rings and support rollers have a radial taper of greater than .015" other kilns may tolerate up to .125" radial taper without noticeable problems. Conical wear is not visible in all but the most severe cases, however, if riding rings thrust hard against retaining blocks/rings, the axial thrust of kiln is not manageable and certain support roller bearings overheat on occasion, it would be best to measure the diameters of the riding rings and support rollers. An electronic measuring instrument can be used during operation, but greater accuracy is achieved by using a pie-tape during a kiln outage. Uneven wear on riding rings and support rollers. Uneven wear is typical of product contamination or erratic axial thrust conditions of the kiln. The adverse affect of uneven wear is it will restrict the ability to adjust support rollers with minimal moves and may affect the axial thrust of the kiln.

2.2.5.Horizontal/diagonal wears on riding rings or support rollers

Horizontal or diagonal wear patterns develop on the surfaces of riding rings or support rollers because of support roller miss-adjustments. If the support roller shaft’s rotational axis is not parallel to each other on each riding ring station, a slip will occur between the riding ring and support rollers and these marks will appear. In severe cases, the marks will cause rough operation of the kiln and high vibration will be noticeable.

2.2.6.Surface spalling on riding rings or support rollers

Surface spalling on the riding rings or support rollers is cause by support roller slope problems, kiln shell misalignment or severe maladjustment of support rollers. If the spalling is severe, stress cracks may form that can weaken the structural stability of the riding rings or support rollers.

2.2.7.Broken edges on riding ring and support rollers

As excessive concave wear develops on the surfaces of support rollers or in rare cases riding rings, the edges will begin to break off as a result of the contact between the components. Again these edges may lead to surface stress cracks that can affect the structural stability of the component parts.

Wear on riding ring thrust face and thrust rollers

In cases where the axial kiln thrust runs in the same direction for extended periods of time, (for example uphill or downhill), the surfaces of the thrust face on the riding ring and the thrust rollers will wear. Excessive wear of these components will cause broken edges on both the riding ring and the thrust roller. Also, if the wear occurs over an extended period of time, the kiln can actually move axially in that direction as the surfaces wear, causing a misalignment between the riding ring and support rollers and the drive gear and pinion.

- Slide bearings of support rollers
- Gear and pinion contact surfaces
- Drive unit overloading
- Retaining rings

2.2.8.Tire and kiln shell interaction

[4]The tire ID is larger than the shell OD. As the kiln rotates, there is circumferential displacement between the tire and the shell as a result. This motion is commonly referred to as tire creep. If the plane of the tire is not perpendicular to the rotating axis of the shell, the direction of motion of the shell is slightly different than the direction of motion of the tire. The tire moves uphill or downhill as a result, until it hits the tire retainers. On other words, the tire creep has a horizontal component. If the angle between the tire plane and the shell is not between 89.5 and 90.5 degrees, the pressure on the tire retainers will be high. Furthermore, since the tire retainer pressure comes from the interaction between the tire ID and the support pads, the rate of wear of the support pads (due to slippage in the horizontal direction) will be high. There are two mechanisms that tilt the tire plane relative to the shell axis:

a) A taper condition on the tire or the support rollers

b) Support roller slopes.

Correcting a high stop block load condition therefore requires either the resurfacing of the tire and the rollers or changing the support roller slopes. Since the tire retainer pressure is caused by a motion component between the tire and the shell, the pressure is directly proportional to the tire creep, and therefore to the tire clearance. Decreasing the tire clearance by 50% (by shimming or replacing the support pads) will decrease the tire retainer pressure by 50%. The source of the force causing retainer pressure is traction between the tire ID and the shell. If the tire ID is lubricated, the coefficient of friction between the tire and the shell will decrease, and the tire retainer pressure will also decrease. The most effective lubricant for this application consists of soft metal powders dispersed in a carrier that evaporates. The lubricant is available in the form of wax bars or a sprayable liquid. The sprayable form prevents the close proximity to moving parts that is required to install the wax bars between support pads. Please note that graphite is a very ineffective lubricant for this application. It does not laminate onto the tire ID, and is very easily dislodged from the surfaces it is supposed to lubricate. Bearing adjustments are commonly made with the intention of decreasing tire retainer pressure. It is mistakenly believed that if, for example, the downhill stop blocks on a pier are overloaded, and the pressure can be decreased by adjusting the support rollers to push the tire uphill. This attempt to fix the problem rests on a lack of understanding of the relationship between two independent sets of forces effecting mechanical stability at each pier. The horizontal motion component between the tire and the roller affects the axial motion of the entire kiln, but will not affect the position of the tire relative to the stop blocks. Stop block pressure is a function of the horizontal motion component between the tire ID and the shell; this motion component is independent of the tire to roller interaction. (For the record, the previous statement is not an absolute truth. Very extreme bearing adjustments, with one roller pushing uphill and the other downhill, will tilt the tire plane relative to the kiln axis, and will thus affect the tire retainer pressure. The magnitude of bearing adjustment required to do this, however, is so extreme that it results in the overload of the support roller bearing thrust mechanism. While this procedure can be beneficial as a short term fix, it does not work in the long term).

2.2.9.Pinion and main gear interaction

[4]The smooth transfer of load from the driving pinion tooth to the following tooth is assured by having two adjacent teeth share the load for a period of time. Tooth A carries the load; tooth B is engaged before tooth A disengages. This load overlap requires that the gear set have the proper root clearance. In other words, if the pinion is set too far out of mesh, the load overlap will be lost. This condition is very typical. As part of the pinion alignment procedure, the pitch lines are usually set apart, in anticipation of a decrease in root clearance as the kiln heats up. Counterintuitive as it may seem, the gears actually separate as a result of thermal expansion. As the kiln diameter expands, the 12 o'clock position of the gear moves up however much the diameter increases; the 6 o'clock position of the shell does not change; the shell axis moves up one half the shell diameter increases. Since the gear is centered on the shell axis, it is lifted out of mesh when the shell axis elevation increases. When setting the pinion on a gear set, the pitch lines should overlap 1/16". As the kiln heats up, the pitch lines will separate via the above thermal mechanism, but the overlap of the gear set will remain adequate.

2.2.10. Roller and tire contact surface condition

[4]The mechanical stability of the tire to roller interaction requires that the roller axis and the kiln rotation axis be parallel. The direction of motion of a rotating object is perpendicular to its axis. If the roller and kiln axis are not parallel, the direction of motion of the two components will not be along the same line. In other words, the motion of the tire relative to the roller will have a horizontal component. This condition causes the kiln to move uphill or downhill and the roller to move in the opposite direction. If the horizontal motion component is high, the roller bearing thrust mechanism overheats from high pressure, and the load on the kiln thrust roller may be high. Correcting this condition requires the adjustment of one of the roller bearings such that the direction of motion of the kiln and the roller become identical.

2.2.11. Bearing thrust mechanism loads

[4]The thrust load on a support roller moves the roller along a line parallel to the kiln axis. The motion of the roller is stopped by a thrust mechanism in one of the bearings. There are two types of bearing thrust mechanisms: a) a heavy bearing housing end plate that contacts the end of the roller shaft and b) a thrust collar or flange on a roller shaft that contacts a flange on the bearing liner. If the thrust load on the roller is high enough, the pressure per square inch on the loaded thrust mechanism will be excessive, resulting in a) lubricant breakdown and bearing failure or b) structural failure of the thrust wear plate. Measuring and minimizing the thrust load on a support roller can prevent these undesirable consequences. Measuring the bearing thrust pressures and making the proper bearing adjustments should be part of the scope of work of a comprehensive hot kiln alignment. The pressure on the thrust mechanism has to be less than 300 psi to assure reliable stability.

2.2.12. Roller shaft to bearing liner interface

[4]The interaction between a roller shaft and a bearing liner will generate high temperature leading to bearing failure unless certain stability conditions are met. The diameter of the brass liner has to be 0.002” greater than the shaft diameter for every inch of shaft diameter. In other words, a 20” shaft has to have a liner with a 20.040” diameter. If this condition is not met, there will not be enough lubricant between the shaft and the liner, and bearing failure can occur. The viscosity of the bearing lubricant is very important. There are no advantages to using low viscosity oils, i.e., ISO 680 or lower. There are however important advantages to using heavy oils. The viscosity of 680 oil is only about 120 cst at 200 degrees F. In other words, the oil viscosity drops by about 80% when the oil temperature increases by 100 degrees. If the room temperature viscosity of oil is 3200, that oil will have sufficient viscosity to lubricate a bearing at up to 350degrees, (a temperature at which 680-oil will fail). If oil viscosity is adequate at an elevated temperature, a bearing will remain stable at that elevated temperature. Since bearing temperatures can rise for a variety of reasons, it is prudent to use the highest viscosity oil, (consistent with the pour point requirements dictated by weather conditions at plant site). The lowest reliable oil viscosity for kiln bearing application is ISO 1500. The uphill and downhill bearings have to be in line to insure uniform and low bearing pressures on both liners. This requirement is often unmet because the bases under the bearings are twisted. If the uphill and downhill bearing base pedestals are not in the same plane, the two brass liners of a roller will not be in line. The bearing liner load distribution will then be uneven. At the position of maximum bearing liner load the bearing pressure may be excessive and the lubricant viscosity may be inadequate for the pressure. This condition often occurs with bearings that use end caps to contain thrust loads (i.e. Fuller and AC bearings). If an end cap pressure is high enough, the bearing places a sufficient torque on the base plate to twist it. The base edge at the end cap bends down and the bearing tilts in the direction of the bent base plate. As the bearing housing tilts, uniform liner load distribution is lost. Base installations and base repairs must be done consistent with the requirement for uniform brass liner load distribution. Note that if a base plate is twisted, a high viscosity oil is less likely to fail (as a result of areas of high bearing pressures) than a light oil.

2.2.13. Roller lubrication

[4]Lubricating the tire-roller interface with graphite will decrease the roller thrust load, and will decrease the pressure on the bearing thrust mechanism. This is because the source of the thrust load is traction in the horizontal direction between the tire and the roller. The graphite lubricant decreases the traction by reducing the coefficient of friction between the tire and the roller. Note: If a bearing overheats and the cause of the high temperature is high roller thrust load, lubricating the roller face will quickly reduce the bearing temperature.

2.2.14. Proper functioning of the hydraulic thrust device

[4]The axial motion of the kiln is limited by thrust rollers. The position of the thrust rollers relative to the tire should be 1/16” off the six o’clock position of the tire, away from the drive pinion side of the kiln. The direction of motion of a rotating thrust tire at any point on the tire circumference is parallel to the tangent to the tire at that point. At the six o’clock position, the direction of motion of the tire is horizontal. Slightly away from the six o’clock position, the direction of motion has a vertical component; the tire moves up or down, in addition to moving horizontally. A vertical motion component at the point of contact between the tire and the trust roller will push the thrust roller down, or lift it up. The resulting friction will wear the thrust roller and the tire. If a tire pushes a misaligned thrust roller down, vertical load builds up. As the tire moves off the thrust roller the vertical load is relieved and there is vertical slippage between the tire and the roller in the process. In time, this vertical movement creates a slight flat spot on the tire and the thrust roller. These flat spots serve as surface discontinuities, where future vertical thrust loads are relieved. Over time the sizes of the flat spots increase, making the thrust roller unusable. Flat spots on the side of the thrust tire can serve as a source for kiln vibrations.

If a thrust roller is not properly aligned relative to the six o’clock position of the tire, there is a danger of lifting the thrust roller shaft out of its bearing housing, and hitting the tire stop blocks. The load capacity of the roller shaft is significantly diminished if it lifts out of the bearing housing. Most thrust roller shaft failures occur by way of this mechanism. The lubricant in the thrust roller bearings should be ISO 3200 grade. Since oil circulation is not a consideration, the high pour point of the oil is immaterial. The high viscosity of the 3200 oil provides significant separation between the shaft and the brass liner.

2.3. Axial Motion of skewed rollers

[1]The primary cause of axial motion of the wheel set is skewing. The flanges, which provide a kinetic support for the wheel set, are always on the inside of the wheels. This produces a wheel cone with its apex outside the wheel set. The effect of these cones is to provide a kinematic stabilization of the wheel set as it runs on the rails. Were the cones reversed, the wheel sets would oscillate wildly in an axial mode as the train moves down the track (ref. 1).

Two basic causes of roller axial motion are identified in this study. These are skewing and externally applied axial thrust. Both causes can be converted into correcting mechanisms that will enable the rollers to roll true (i.e., in their plane of contact without axial motion). The first correcting mechanism is kinematic; the second is kinetic. Although a kinetic or force correction, in the form of a roller flange or bumper, is sometimes required to insure that gross axial motion cannot occur, the kinematic correction is unquestionably more desirable. The kinematic correction, which results in self-centering action, can be implemented without seriously affecting either the mechanical efficiency or durability of the contact. Axial motion that is not corrected kinematically will create significantly large thrust forces that have to be withheld by flanges or thrust bearings. The investigation conducted by Virabov (ref. 7) shows that even a minute misalignment between the contact and spin axes of interacting cylindrical rollers caused by unavoidable errors in the manufacture or alignment of the rolling bodies will result in significant axial forces being generated. Tests results for a pair of 50millimeter (2-in.) diameter steel rollers running dry (traction coefficient = 0.32) indicated that a roller skew angle of only 0.27' will produce a thrust force that is 24 percent of the normal load acting on the body. Obviously, roller flanges or thrust bearings that are required to operate continuously under such conditions would be unattractive from a system life and performance standpoint.

In a private communication to the authors, Dr. Veljko Milenkovic extended his work regarding the stability of a railroad wheel-rail contact (ref. 1)to the more generalized case of roller pairs in free-rolling contact. In this communication, he advocated the technique of using self-corrective geometries to promote roller stability. Milenkovic's

2.4. Conclusion

So far we have seen that rotary kiln axial thrust can be controlled in an individual approach. However, this approach could take longer time to adjust any misalignment after observing the effect of one trial. This thesis is supposed to solve the aforementioned problem by applying an integrated approach for both skewing angle and rotational speed of the rotary kiln with the possible values of current variables fed. And this could show where exactly the problem lies so that the action to be taken wouldn’t be as such longer than hours.

CHAPTER THREE

3. MATERIAL, METHOD, CONDITION

3.1. MATERIAL

Even though the problem selected to address comprises possibly all rotary kilns that are engaged to manufacture mining products, materials for this thesis are decided to collect from Muger Cement Factory rotary cement kiln. All data necessary to carry out the thesis will be utilized from MCF existing line. It has been tried to indicate that the objective of the thesis is to contribute for the formulation of a set up in control mechanism for rotary kiln axial thrust. The effect of rotational speed plays an important role in the magnitude of the thrust that exerts on different components of kiln system. Data are to be collected for a random capacity of rotary kiln which is for this particular thesis Mugher cement factory kiln No. one has been considered. These data include kiln length and diameter, number of base-rollers, rotational speed of the kiln, contact surface condition between tire and rollers, slope of rotary kiln axis from design point of view etc…

3.2. DIMENSION

[5]All data which are related to the study of the rotational speed have been indicated in the table below.

Figure 1. Mugher Cement Factory kiln no. I design parameters

Data No.	Description of the data		Measuring unit	QNT/Size /Amount
1	Kiln power requirement		Kw	180
2	Kiln speed	Maximum	RPM	2.3
		Minimum	RPM	0.57
3	Girth gear	Number of teeth	Pcs	152
		Module	-	40
		Teeth width	mm	500

Data No.	Description of the data		Measuring unit	QNT/Size /Amount
4	Pinion	Number of teeth	Pcs	27
		Module	-	40
		Teeth width	mm	550
5	Girth gear to pinion transmission ratio		-	5.63
6	Main gearbox power		Kw	180
7	Driving speed	Maximum	RPM	730
		Minimum	RPM	182
	Output speed	Maximum	RPM	13
		Minimum	RPM	3.25
8	Transmission Ratio		-	56
9	Total mass of the rotary kiln		Ton	694.6
10	Rotary kiln length		m	65
11	Diameter of kiln		m	4.2
12	Slope of rotary kiln		%	3

3.3. METHOD

As earlier in chapter one tried to explain that the scope of the thesis is limited to exploring the effect of varying speed of a rotating kiln and skewing angle of support rollers as among many they are the factors affecting axial thrust, the appropriate method has been taken as **Mathematical modeling and ANSYS software**. The mathematical modeling includes static and dynamics. Mathematical modeling is supposed to help generate inputs for ANSYS software. Because capability of version of ANSYS software and processing speed of PC I have is under question the complex dynamic analysis of rotary kiln system using ANSYS software has been found so difficult. Hence static analysis is engaged in the ANSYS software to show the effect of rotational speed on axial thrust. It is assumed that ANSYS will show indirectly tendency of axial movement if there is an increase in directional displacement along Z-axis as the tangential load changes due to change in rotational speed. Besides number of components and their dimensions in the system are extremely huge as compared to the equipment (PC) capacity to solve, only the middle tire and a pair of support rollers are taken to model this phenomenon. That is why mathematically static analysis has been used to show which of the three pairs of support rollers carry the maximum normal load. This is because it is assumed that if we make a certain skewing angle on this pair of support rollers effective axial movement could be observed for the fact that axial load would have a direct relation with normal load of the carrying roller. Mathematical analysis is also another method that is engaged to find the tangential force. Here, by varying angular speed of the rotary kiln we can get different values of tangential force. The resultant of these two forces (i.e. normal load on the support roller and rotation-causing tangential force) will have an axial effect for a chosen value of skewing angle. Assuming that there is enough traction-causing frictional force between tire and support rollers the idea of helical gear meshing that causes an axial load due to helix angle has been taken for this analysis. Helix angle is analogous to skewing angle. The possible methods that will be engaged for future investigation of axial thrust affecting factors are that the dynamics together with contact surface modeling is supposed to fit for Tire and kiln shell interaction, roller and tire contact surface condition, bearing thrust mechanism loads, hydraulic thrust device, pure contact surface model analysis is to be employed for pinion and main gear interaction, roller shaft to bearing liner interface and roller lubrication. With all these model analysis carried out simulation of the effect for different values of all parameters will be done.

3.4. ANALYSIS AND RESULT

3.4.1. Static Analysis for Mass of Kiln System

Static analysis needs to be carried out in order to locate the maximum normal load that the rotary kiln exerts so that we can identify which of the three pairs of base rollers accommodates lion's share of the pressure load. This is done because axial movement of the rotary kiln is caused by the traction force that is created due to the interaction between the pairs of rollers and tire ring because of their non parallel axes or skewing.

The following points are to be considered when mass of a running kiln shell system is supposed to determine

1. Thinking of mass of the whole body to be concentrated at a point along the axis and calculate for center of mass
2. Considering thickness differences of segmented shell bodies and calculate for masses of each segmented body, tire and girth gear mass
3. Considering types of bricks lined at different zones of kiln shell to determine specific densities so that one can calculate mass of lining bricks for each zone

The components that result in the whole mass system of a rotary kiln include

3.4.1.1. Refractory bricks

[5]More than 60% of the mass system is comprised by refractory bricks that are lined the whole length of rotary kiln. Depending on the process zone on which phenomena of different chemical and physical properties take place, the types of bricks to be lined and the length of lining differ also. It is found that the mass per meter length of bricks lining for upper and lower transition and central burning zone is found to be maximum which is 7493kg/m. This finding will help choose which of the three pairs of support rollers experience high normal load.

Here are the names of zones and respective load shares of refractory bricks

3.4.1.2. Inlet zone

Length of zone is 1.4m. For a single ring, 128 bricks are needed and mass of a single brick is:

$M=3.873dm^3 x1000x2.175kg/cm^3$

$=8.42Kg$

Mass of a ring of bricks=8.42kgx128bricks

=1078.2kg

Mass of bricks in one meter per ring=1078.2kg/ringx5ring/meter

=5391.2kg/meter

Mass of bricks in inlet zone=5391.2kg/meterx1.4m

=**7547.7kg**

3.4.1.3. Calcining zone

Length of zone is 28.76m. For a single ring, 128 bricks are needed and mass of a single brick is:

$M=3.873dm^3 x1000x2.175kg/cm^3$

$=8.42Kg$

Mass of a ring of bricks=8.42kgx128bricks

=1078.2kg

Mass of bricks in one meter per ring=1078.2kg/ringx5ring/meter

=5391.2kg/meter

Mass of bricks in calcining zone=5391.2kg/meterx28.76m

=**155051.4kg**

3.4.1.4. Safety zone

Length of zone is 8.8m. For a single ring, 128 bricks are needed and mass of a single brick is:

$M=3.873dm^3 x1000x2.675kg/cm^3$

$=10.36Kg$

Mass of a ring of bricks=10.36kgx128bricks

=1326.11kg

Mass of bricks in one meter per ring=1326.11kg/ringx5ring/meter

=6630.57kg/meter

Mass of bricks in safety zone=6630.57kg/meterx8.8m

=**58349kg**

3.4.1.5. Upper transition zone, Lower transition zone and Central burning zone

These three zones can be considered to use the same type of bricks which is called basic and have densities from 2.95 kg/cm^3 to 3.10kg/cm^3. Hence total length of the three zones:

Total length=4m+13m+6m

=23m

Length of zone is 23m. For a single ring, 175 bricks are needed and mass of a single brick is:

M=2.831dm^{3}x1000x3.025kg/cm^3

=8.564Kg

Mass of a ring of bricks=8.564kgx175bricks

=1498.66kg

Mass of bricks in one meter per ring=1498.66kg/ringx5ring/meter

=7493.3kg/meter

Mass of bricks in the three zones=7493.3kg/meterx23m

=**172345kg**

3.4.1.6. Cooling zone

Length of zone is 3m. For a single ring, 128 bricks are needed and mass of a single brick is:

M=3.873dm^{3}x1000x2.825kg/cm^3

=10.94Kg

Mass of a ring of bricks=10.94kgx128bricks

=1400.5kg

Mass of bricks in one meter per ring=1400.5kg/ringx5ring/meter

=7002.4kg/meter

Mass of bricks in cooling zone=7002.4kg/meterx3m

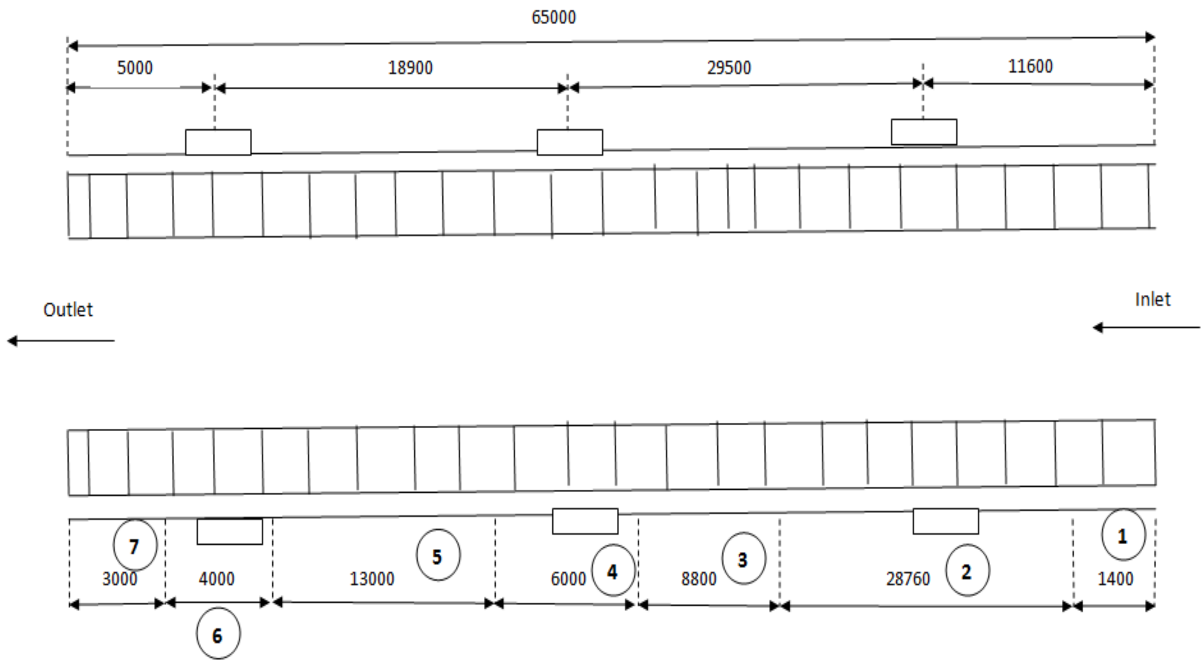
=**21007.2kg**

Figure 2. Summary of mass Analysis for bricks at different zones of kiln

Zone type	Zone length(meter)	Bricks type	Volume(dm^3)	Density(g/cm^3)	No. of bricks per ring	No. of rings per meter	No. of bricks per meter	No. of bricks in zone	Mass of bricks per piece	Total mass of bricks in zones (Kg)
Inlet	1.4	KROLEX 40	3.873	2.175	128	5	640	896	8.42	7544
Calcining	28.76	KROLEX 40	3.873	2.175	128	5	640	1840	8.42	154982
Safety	8.8	KROLEX 70	3.873	2.675	128	5	640	5632	10.36	58348
Upper transition	6	PERILEX83	2.831	3.025	175	5	875	5250	8.564	44961
Central burning	13	PERILEX83	2.831	3.025	175	5	875	11375	8.564	97416
Lower transition	4	PERILEX83	2.831	3.025	175	5	875	3500	8.564	29974
Cooling	3	KROLEX 85	3.873	2.825	128	5	640	1920	10.94	21005
Total Mass of bricks										**414230**

Figure 3. Rotary kiln zones

1	2	3	4	5	6	7
Inlet	Calcining	Safety	Upper transition	Central burning	Lower transition	Cooling



3.4.1.7. Rotary kiln shell with different thickness for different zones

The sheet metal shell has an inside diameter of 4.2m and an overall length of 65m measured from the inlet to the outlet end. Sheet metal of different thickness will be used for the individual zones depending on the calculated loads and on the thermal influences. Based on physical measurement taken during maintenance period the thickness of section of the shell for upper transition zone, central burning zone and lower transition zone is found 40mm whereas the rest zones have shell thickness of 30mm. Keeping the above information in mind we can calculate the mass of the shell on zone bases and compare load sharing phenomenon for the paired support rollers.

Section I

Mass of Kiln shell=$(\pi/4)(4.24^2-4.2^2)$x7850kg/m^{3}x23m

=**47872.9kg**

Section II

Mass of Kiln shell=$(\pi/4)(4.23^2-4.2^2)$x7850kg/m^{3}x42m

=**65425kg**

Where

Density of steel sheet to be 7850kg/m^3

As a result of this procedure mass per meter length of kiln shell has been found to be big(2081kg/m) for the three zones as compared to the rest ones(1559kg/m). Hence a total mass of kiln shell has been calculated as 113298kg.

3.4.1.8. Tire rings at three basements

Three cast steel tires are mounted on the kiln shell so that a favorable load distribution is obtained.

Figure 4. Kiln tire dimensions including mass

No.	Position from inlet (mm)	Outer Diameter (mm)	Inside Diameter (mm)	Mass (kg)
Tire I	11600	5080	4356	31600
Tire II	41100	5100	4376	38100
Tire III	60000	5080	4356	32900

3.4.2.Determining Center of Mass

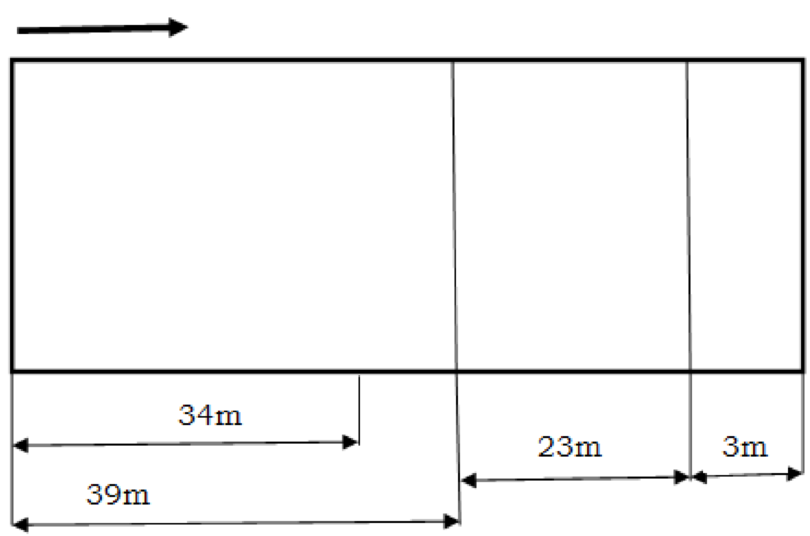
3.4.2.1. Mass center of the kiln shell:

113C=19.48x60747.35+50x47872.9+63.46x4677.67

C=34m

Kiln shell

Figure 4a. Mass center of kiln shell from inlet end



3.4.2.2. Mass center of refractory bricks

3.4.2.2.1. Inlet zone

The center of mass for inlet zone refractory bricks as counted from the inlet end is calculated as the length of bricks lining divided by two which is:

$C_I = L_I/2$

$= \mathbf{0.7m.}$

3.4.2.2.2. Calcining zone

The center of mass for calcining zone refractory bricks is calculated as the sum of the lengths for inlet zone and half the length of the calcining zone which is:

$C_c = L_I + L_c/2$

$= 1.4 + 14.38$

$= \mathbf{15.08m}$

3.4.2.2.3. Safety zone

The center of mass for safety zone refractory bricks is calculated as the sum of the lengths for inlet zone, calcining zone and half the length of the safety zone which is:

$C_s = L_I + L_c + L_{s/2}$

$= 1.4m + 28.76m + 8.8/2m$

$= \mathbf{34.56m}$

3.4.2.2.4. Upper transition, Central and Lower transition zone

Since the types of bricks to be lined on these zones have identical physical and chemical properties one can calculate the center of mass as one zone. Hence it is as the sum of the lengths for inlet zone, calcining zone, safety zone and half the combined lengths of the three zones:

$C_b=L_I+L_c+L_s+L_b/2$

$=1.4m+28.76m+8.8m+23/2m$

$=$**50.46m**

3.4.2.2.5. Cooling zone

The center of mass for cooling zone refractory bricks is calculated as the sum of the lengths for inlet zone, calcining zone, safety zone, combined burning zone and half the length of the cooling zone which is:

$C_{cl}=L_I+L_c+L_s+L_b+L_{cl}/2$

$=1.4m+28.76m+8.8m+23m+3/2m$

$=$**63.46m**

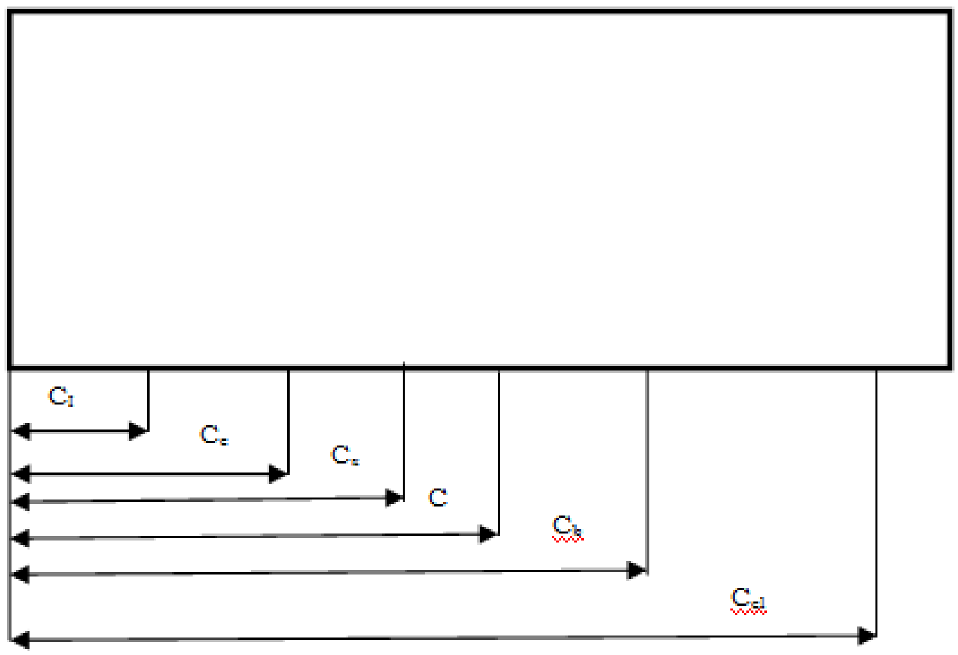
Finally the mass center of refractory bricks can be calculated as:

$Cx414=C_Ix7.547+C_cx155+C_sx58.35+C_bx172.3+C_{cl}x21$

$Cx414=0.7x7.547m+15.08x155m+34.56x58.35m+50.46x172.3m+63.46x21m$

$C=$**35.13m**

Figure 4b. Bricks zone center masses



Where:

C_I-Center of mass for inlet zone bricks lining

C_c-Center of mass for calcining zone bricks lining

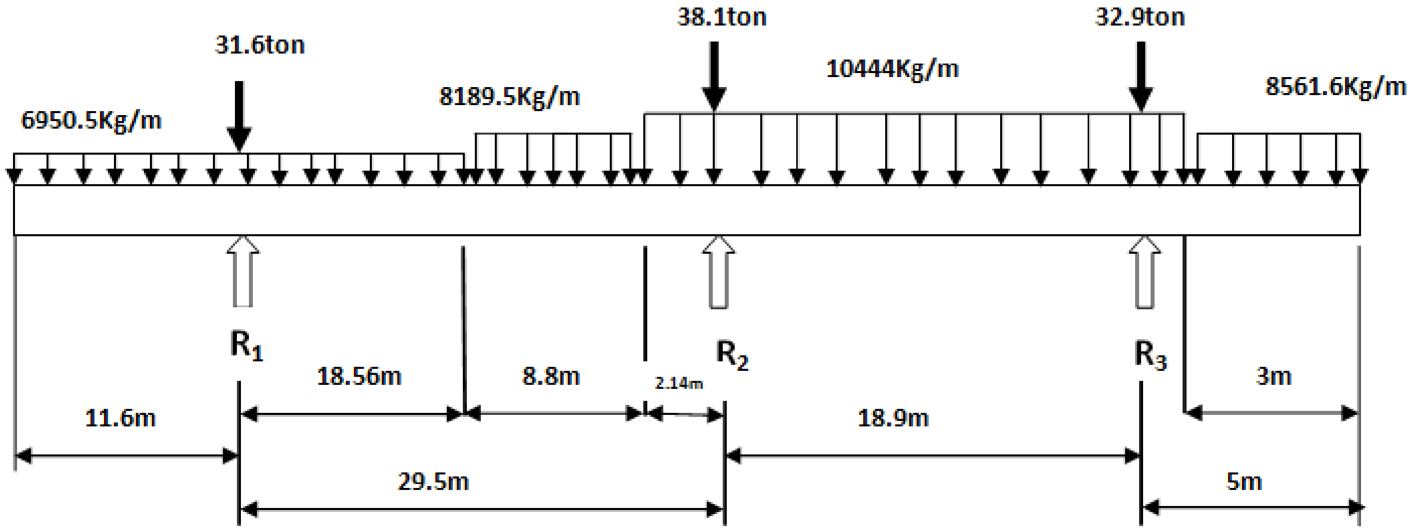
C_s- Center of mass for safety zone bricks lining

C-Center of mass for whole combined zones of bricks lining

C_b-Center of mass for burning zone bricks lining

C_{cl}-Center of mass for cooling zone bricks lining

Figure 4c. Static modeling of the kiln system



Applying equilibrium conditions

$\sum_1^3 F_y = 0,\ \ R_1 + R_2 + R_3 = 680$ ………………………………a

$\sum M_0 = 0,\ \ 11.6R_1 + 41.1R_2 + 60R_3 = 23336.2$ ……………b

Since the equilibrium condition is not enough to solve for three unknowns we need to incorporate superposition method which is used for redundant reactions.

Hence the sum of deflections from all loads and reactions at a point on the span of the beam results in zero effect.

$\sum \nabla_{p_1}$=0

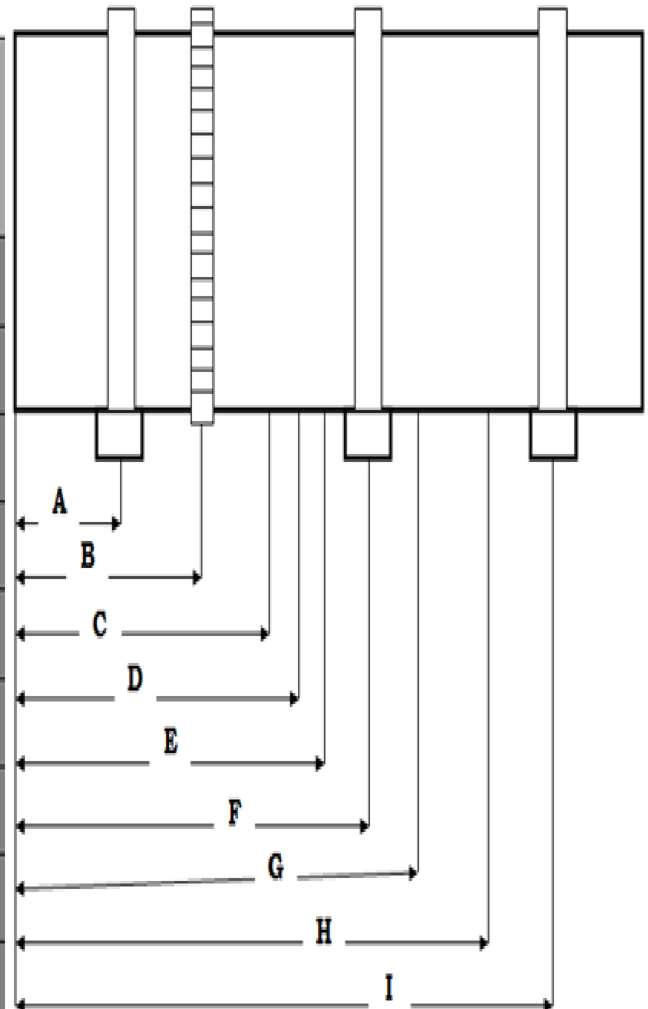
$342.2R_2 + 561.4R_3 = 184694.48$ ………………………….c

Figure 5. Results of Analysis for reaction forces on the support rollers

Basement	R1	R2	R3
Load(Ton)	188	440	67

Figure 6. Results of Analysis for overall Center of mass of kiln system

Letter	Center of mass type	Center of mass (Meter)	Quantity of mass (Ton)
A	First tire	11.6	31.6
B	Girth gear	16.1	30
C	Kiln shell	34	113
D	Kiln mass system	34.9	694.6
E	Bricks lining	35.13	414
F	Second tire	41.1	38.1
G	Coating	42	15
H	Material flowing	43	20
I	Third tire	60	32.9



3.4.3.Dynamic Analysis

[5]It is important to understand that traction force depends on the normal force applied at contact surfaces of the two rolling components. There are hence different options to examine axial movement of rotary kiln by skewing optionally three pairs of support rollers either doing for all pair together or individual treatment. In this case we have considered starting on the support roller pair with maximum normal load shared.

Determining component of the traction force along the kiln axis for a pair of support rollers at a maximum pressure load will help find out displacement of rotary kiln along its axis. It's also important to check up if speed variation of the rotary kiln affects traction force and thereby rate of displacement of kiln along its axis. In addition to the static load, rotation of the kiln has an influence on the traction force due to the skewing of support rollers. Hence torque has to be analyzed. Torque is applied by the drive set of the rotary kiln. This torque is transmitted through the pinion-girth gear meshing mechanism to the mega mass of rotary kiln system. The kiln rolls on three pairs of support rollers. The roller and tire contact surface generates traction force for the kiln to move axially upward or downward depending on the skewing directions of the paired rollers. Skewing refers shifting of the paired roller axes from the tire ring axis i.e. the roller axes and the tire ring axis will become non-parallel and that is said to be roller skewing.

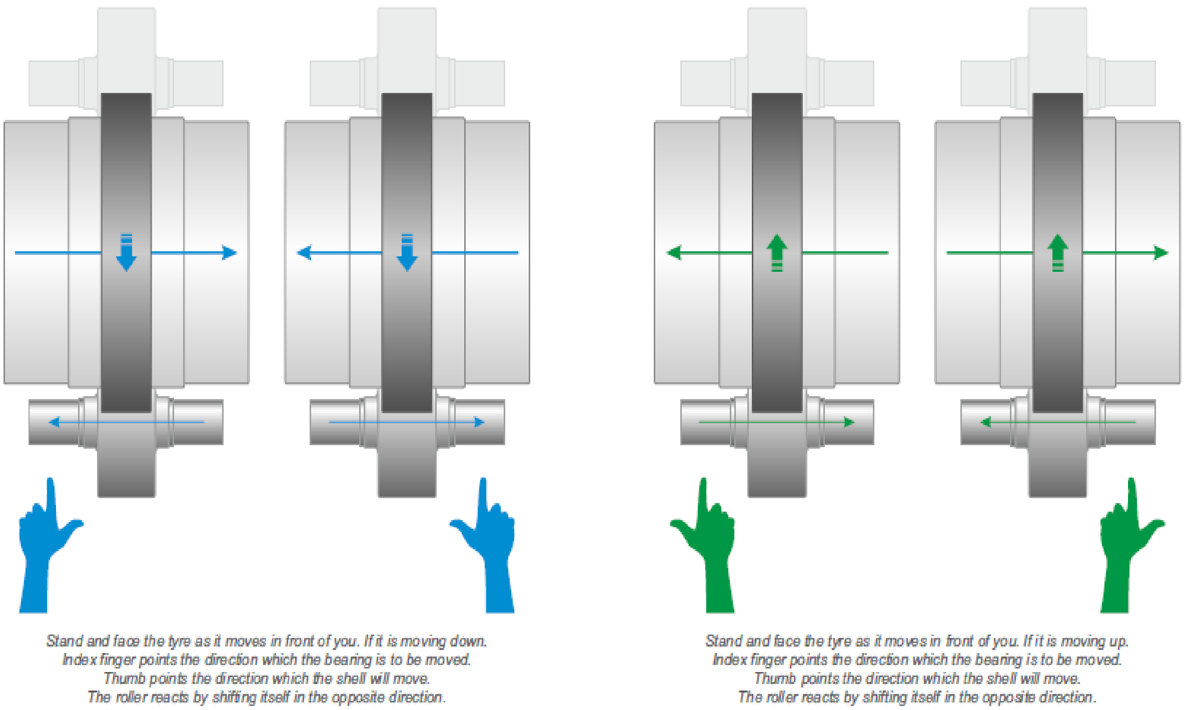
How far the kiln advances upward for a defined skewing value on a single pair of support rollers and pre-determined traction coefficient between tire ring and support rollers in a certain period of time? And it is also important in the analysis to observe the axial movement and its rate for different values of rotational speed of kiln.

Figure 7. Dimensions to be determined at the end

Dimensions to be determined	Rotational speed of kiln(RPM)				Skewing angle of support roller for middle pier			
	1.75	2.0	2.1	2.3	0.21	0.32	0.42	0.57
Axial	0.299	0.299	0.299	0.299	2764.4	3472.4	3873.4	4583.2

Displacement(µm)								

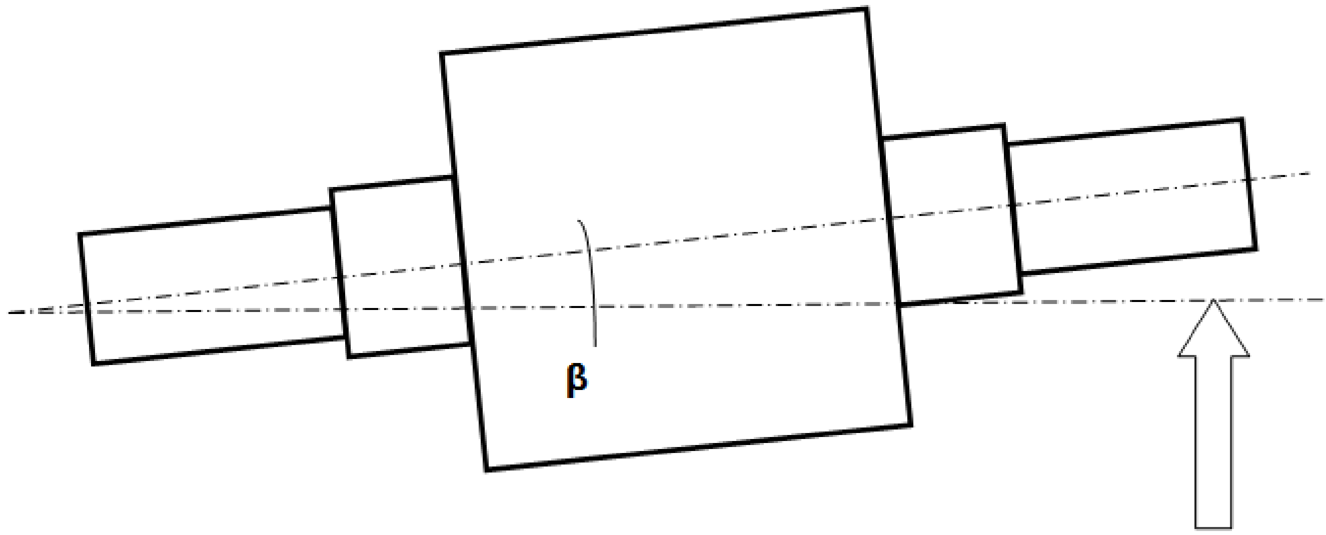
Figure 8. Basic Roller Adjustment rules



3.4.4.Skewing of support roller

Practically skewing is done for many values depending on the desire of rotary kiln alignment from the smallest value (10mm push from one end of support roller shaft) to give a thrust relief for a sliding bearing collar to the largest one (27mm) to initiate an upward axial motion of rotary kiln in the history of Mugher Cement Factory for first line.

Figure 9. Roller skewing as angle to be taken the largest one



Skewing amount (mm)	10	15	20	27
Skewing angle(β)	0.21	0.32	0.42	0.57

3.4.5.Contact of Two Cylinders with Axes Parallel

[2]When any two curved bodies of different radii of curvature are brought into contact they will initially touch at either a point or along a line. With the application of the smallest load, elastic deformation enlarges these into contact areas across which the loads are distributed as pressures

If two circular cylinders with radii R1 and R2 are pressed together by a force per unit length of magnitude P with their axes parallel, as shown in Figure 10, then the contact patch will be of half-width b such that

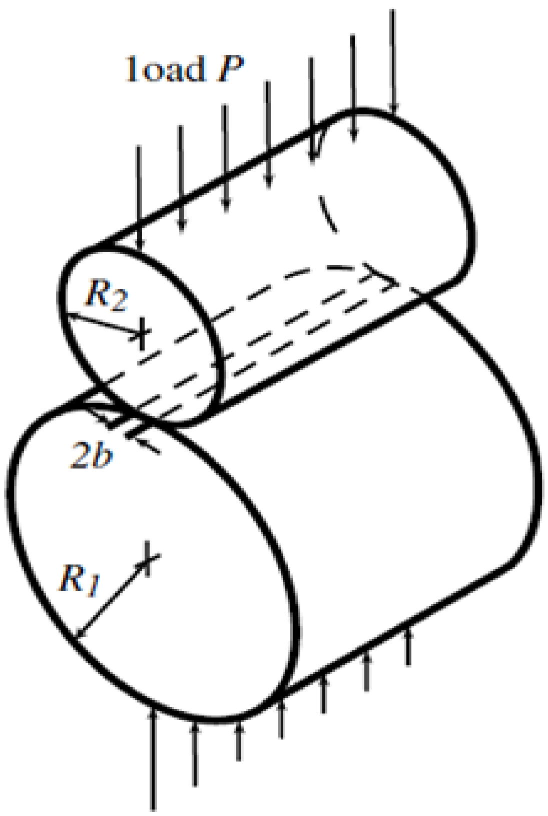
$$b = \left\{\frac{2PR}{\pi E^*}\right\}^{1/2} \quad \ldots\ldots\ldots\ldots 1$$

where R and E* are the reduced radius of contact and the contact modulus defined by

$$\frac{1}{R} = \frac{1}{R_1} + \frac{1}{R_2} \quad \ldots\ldots\ldots\ldots (1a)$$

Figure 10. Parallel cylinders in elastic contact;

the intensity of the loading is P per unit length



and

$$\frac{1}{E^*} = \frac{1-v_1^2}{E_1} + \frac{1-v_2^2}{E_2} \quad \text{...(2)}$$

respectively.

Contact pressure is again semielliptical such that

$$P(x) = P_o\left\{1 - {r^2}/{b^2}\right\}^{1/2} \quad \text{..(3)}$$

where peak pressure

$$P_o = \left\{ {PE^*}/{\pi R} \right\}^{1/2} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (4)$$

and coordinate x is measured in a direction perpendicular to that of the cylinder axes. The mean pressure pm over the contact strip is equal to P/2 b and is given by

$$P_m = {\pi P_o}/{4} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (5)$$

The axes of the cylinders move together by a small distance Δ where

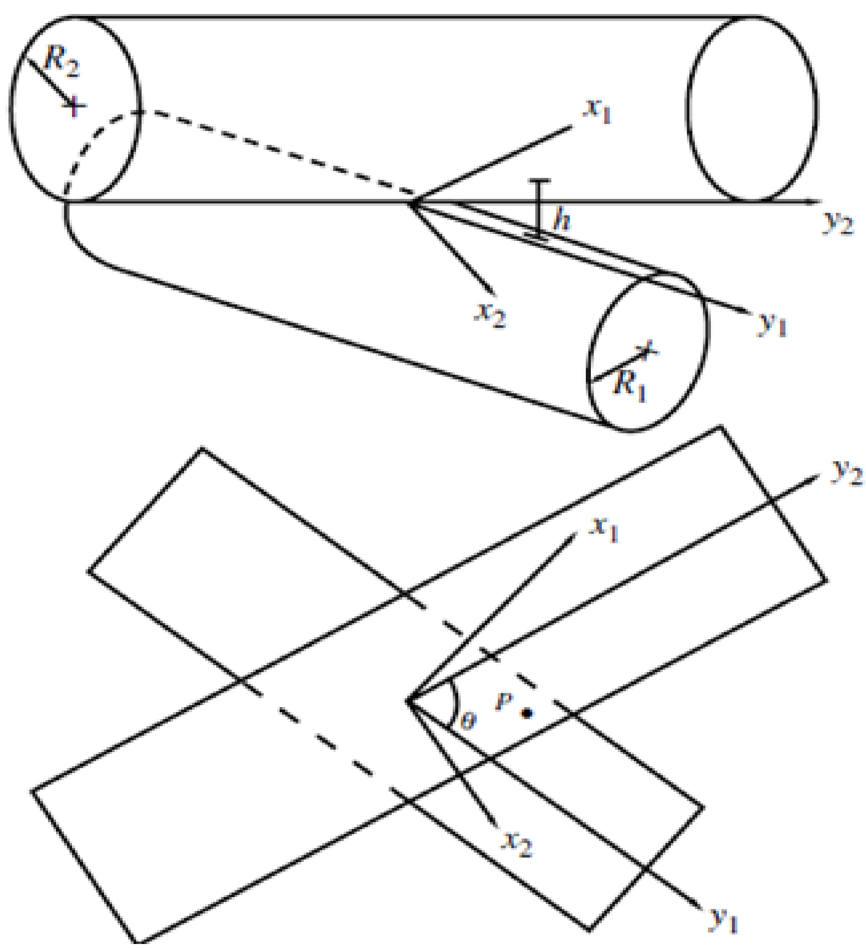
$$\Delta = (1 - v_1^2)\frac{\left[\ln\left({4R_1}/{b}\right) - {1}/{2}\right]}{E_1} + (1 - v_2^2)\frac{\left[\ln\left({4R_2}/{b}\right) - {1}/{2}\right]}{E_2} \quad \ldots\ldots\ldots\ldots\ldots\ldots (6)$$

Once again, if one of the surfaces is planar, then the value of R will be equal to the radius of curvature of the other.

3.4.6.Contact of Two Cylinders with Inclined Axes

[2]Suppose the lower cylinder is of radius R_1 and the upper of R_2, as shown in Figure 11. The cylinders touch at the point O but their axes are inclined at an angle θ. O_{x1y1} is a set of Cartesian axes with O_{y1} along a generator of the lower cylinder, and O_{x2} a second similar set with O_{y2} along a generator of the upper cylinder.

Figure 11. The geometry of two circular cylinders in skewed contact



Close to the origin we can approximate the circular section of each cylinder by a parabolic profile and so write that the separation h of the two solid surfaces at the point P, which has coordinates (x_1,y_1) in the first coordinate system or (x_2,y_2) in the second, is given by

$$h \approx \frac{x_1^2}{2R_1} + \frac{x_2^2}{2R_2} \quad \ldots\ldots\ldots\ldots(7)$$

It is possible to choose a common set of axes (O_{xy}), in which O_x is inclined to the O_{x1}-direction by the angle α, such that the original Heinrich Hertz formula is recovered which is

$$h = Ax^2 + By^2 \quad \ldots\ldots\ldots\ldots(7a)$$

where h is the gap between the undeformed surfaces , x and y are orthogonal coordinates lying in the common tangent plane to the two surfaces, provided that A and B satisfy the conditions

$$B - A = \frac{1}{2}\left\{\frac{1}{R_1^2} + \frac{1}{R_2^2} + \left({}^{2}/_{R_1R_2}\right)\cos 2\theta\right\}^{1/2} \quad \ldots\ldots\ldots\ldots(8)$$

and

$$B + A = \frac{1}{2}\left\{{}^{1}/_{R_1} + {}^{1}/_{R_2}\right\} \quad \ldots\ldots\ldots\ldots(9)$$

And α is given by the solution of

$$\frac{R_2}{R_1}\sin 2\alpha = \sin 2(\theta - \alpha) \quad \ldots\ldots\ldots\ldots(10)$$

Equation (7a) can then be written as

$$h = \frac{x^2}{2R'} + \frac{y^2}{2R''} \quad \ldots\ldots\ldots\ldots(11)$$

by defining

$$R' = {}^{1}/_{2A} \ \text{ and } R'' = {}^{1}/_{2A} \quad \ldots\ldots\ldots\ldots(12)$$

R′ and R″ are known as the principal radii of relative curvature.

It is evident from Equation (12) that contours of constant gap h between the undeformed surfaces will be ellipses, the lengths of whose axes are in the ratio $(R'/R'')^{1/2}$.

When a normal load P is applied, the point of contact spreads into an elliptical area with semi-axes a and b, such that the eccentricity, i.e., the ratio b/a, is independent of the load and depends only on the ratio of R′/R″. For mildly elliptical contacts, i.e., A/B < 5, then the ratio b/a is given by

$$b/a \approx (A/B)^{2/3} \quad \text{...}(13)$$

An "equivalent" radius R_e can be defined as

$$R_e = (R'' \times R')^{1/2} = \frac{1}{2}(AB)^{-1/2} \quad \text{..............................}(14)$$

and this used to estimate the contact area or Hertz stress by using the circular contact equations with R replaced by Re. The approach of the bodies can be calculated using equation

$$\Delta = a^2/R = a\pi P_o/2E^* = \left\{9P^2/16RE^{*2}\right\}^{1/3} \quad \text{.................}(14a)$$

but with R replaced by $(AB)^{-1/2}$ rather than Re.

If the two cylinders make contact with their axes parallel so that

θ = 0, it follows from Equations 8, 9, and 12 that

$$\frac{1}{R'} = \frac{1}{R_1} + \frac{1}{R_2} \quad \text{................}(15)$$

i.e. Equation 2 is recovered.

On the other hand, if the axes of the two cylinders are perpendicular to one another θ = 90°; then $R' = R_1$ and $R'' = R_2$. It follows that

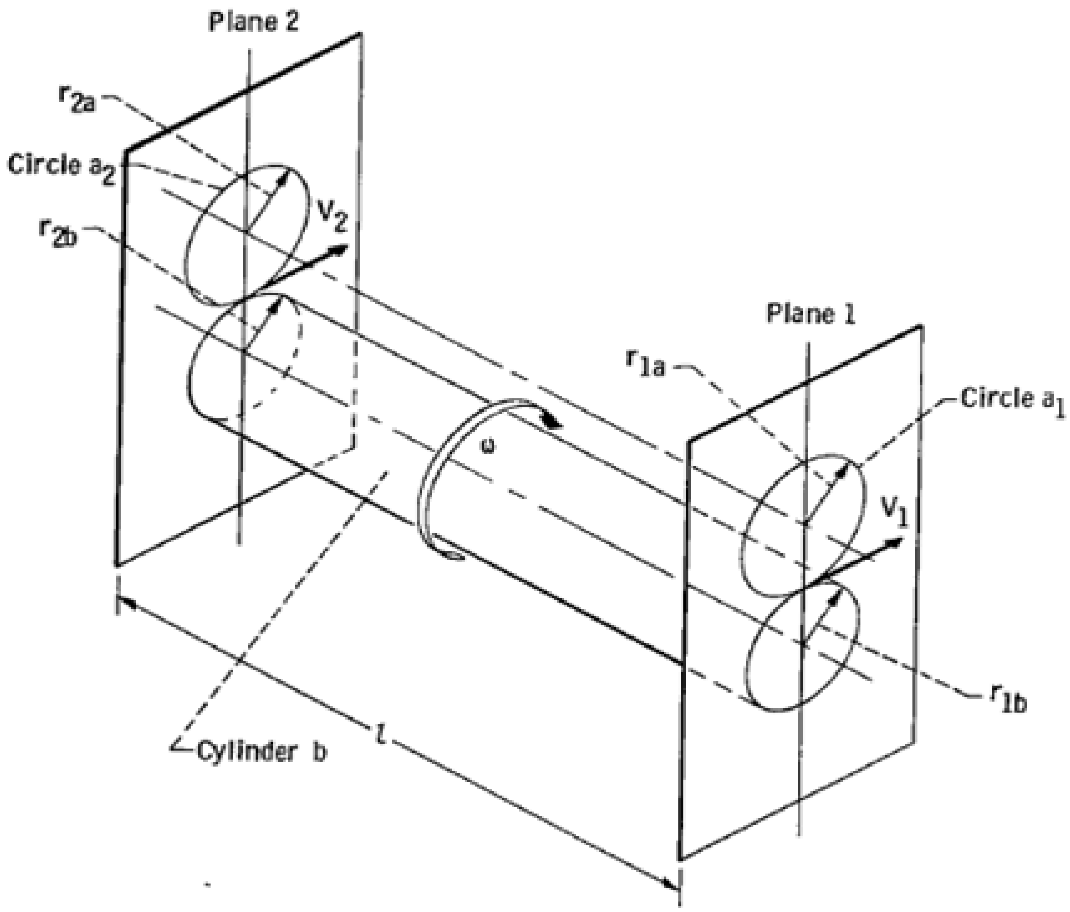
$$h = \frac{x^2}{2R_1} + \frac{y^2}{2R_2} \quad \text{................}(16)$$

Consequently, for the particular case of a pair of equal cylinders crossing at an angle of 90°, contours of constant surface separation close at the contact point will be circles.

3.4.7.Axial Motion

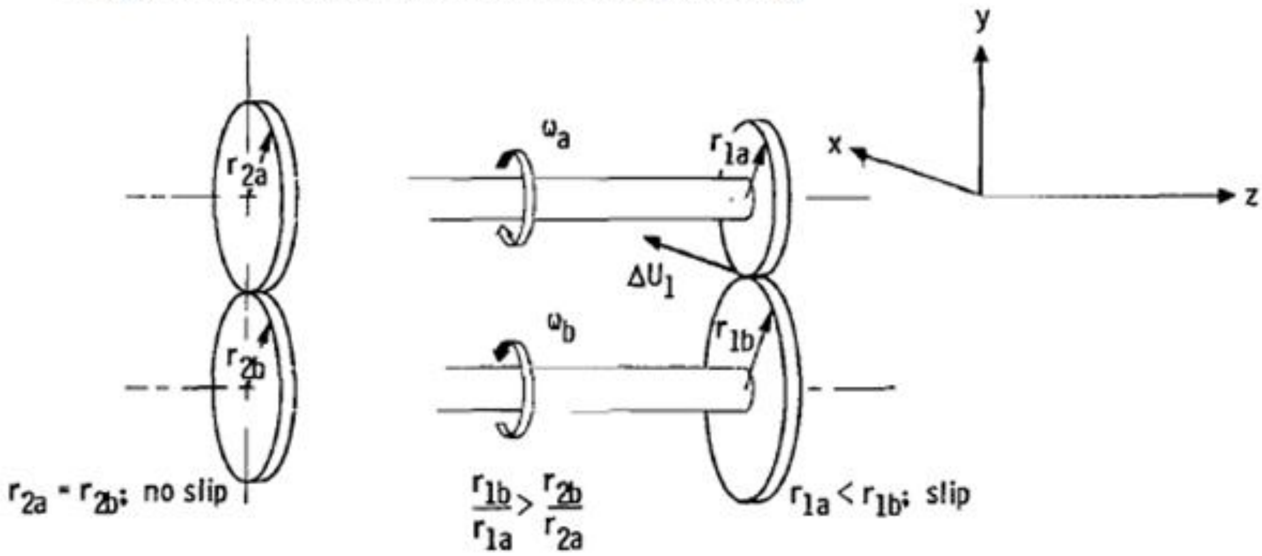
[1]The contact of interest is that of two cylindrical elements rolling on each other. This can be modeled in one plane as a pair of rolling circles or in a continuous series of parallel planes as a sequence of rolling circles rigidly attached to each other along the respective axes of the two cylinders. A kinematic analysis of the effects of misalignment requires that at least two planes of contact be considered, such as the two planar models shown in the figure below.

Figure 12. Two-plane rolling model

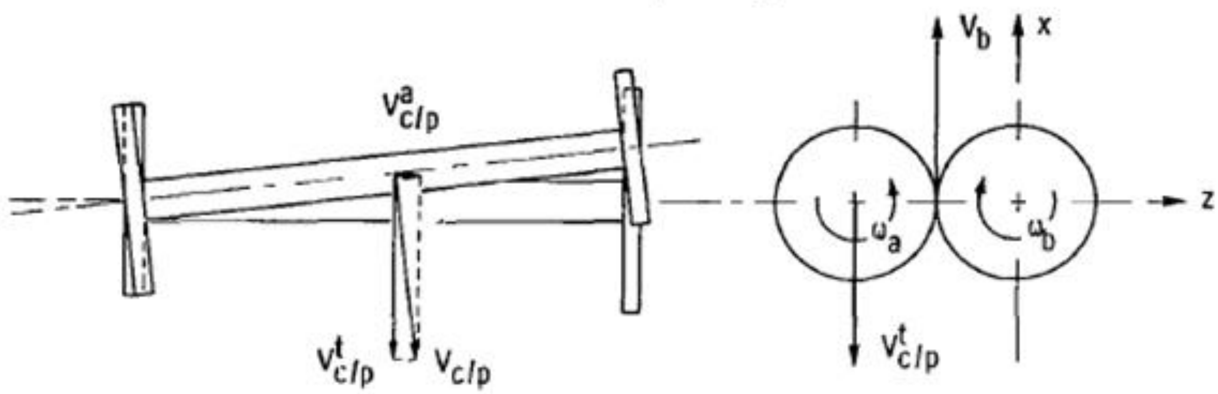


In the model slight variations in rolling radii from the nominal radii r_a and r_b are considered. However, each cylinder is a rigid body, thus the planar rolling circles at each end of the cylinder must move as a single unit. In plane 1 the pitch point velocity of cylinder b is given by

Figure12a Roller kinematic instability



(a) Differential slip velocity.



(b) Skewing and axial velocity generation.

$$V_{1b} = R_{1b}\omega_b \quad \text{.................}(17)$$

while in plane 2, it is

$$V_{2b} = R_{2b}\omega_b \quad \text{.................}(18)$$

For cylinder a these two velocities would be

$$V_{1a} = R_{1a}\omega_a \quad \text{.................}(19)$$

and

$$V_{2a} = R_{2a}\omega_a \quad \text{.................}(20)$$

For no slip at plane 2

$$V_{2a} = V_{2b} \quad \text{........................}(21)$$

or

$$R_{2a}\omega_a = R_{2b}\omega_b \quad \text{.................}(22)$$

$$\omega_a = \frac{R_{2b}}{R_{2a}}\omega_b \quad \text{........................}(23)$$

In plane 1

$$\frac{V_{1a}}{R_{1a}} = \frac{R_{2b}}{R_{2a}}\omega_b \quad \text{.................}(24)$$

For no slippage at plane1

$$V_{1b} = V_{1a} \quad \text{.....................}(25)$$

From the above equations

$$\frac{R_{1b}}{R_{1a}}\omega_b = \frac{R_{2b}}{R_{2a}}\omega_b \quad \text{.................}(26)$$

or

$$\frac{\mathrm{R_{1b}}}{\mathrm{R_{1a}}} = \frac{\mathrm{R_{2b}}}{\mathrm{R_{2a}}} \quad \text{.......................}(27)$$

As shown in figure 2(a), if r_{lb}/r_{la} is greater than r_{2b}/r_{2a} and pure rolling exists at the contact in plane 2, then slippage will occur at the contact in plane 1. For the geometry shown, where $r_{2a} = r_{2b}$ and $r_{la} < r_{lb}$, a slip velocity will be developed in the positive x direction between the contacting points on the roller surfaces. Thus,

$$\Delta U_1 = V_{1b} - V_{1a} \quad \text{.................}(28)$$

will be a positive slip velocity of roller b under roller a. Friction will convert this relative velocity into a tractive force that tends to skew cylinder a on cylinder b. This skewing will generate a component of the relative velocity of cylinder a's center with respect to the pitch point in the axial direction from plane 2 to plane 1. This velocity component is in the direction towards the smaller rolling radius circle. The tangential component of this relative velocity $Vt_{c/P}$ is subtracted from the pitch line velocity V for the net circumferential motion. The axial component of this relative velocity $Va_{c/P}$ remains unopposed. This produces a net axial motion of cylinder a with respect to cylinder b. Unopposed, this axial motion or 'walking", which a consequence of roller skewing, will continue until either a mechanical restraint is encountered or the rollers have become disengaged.

3.4.8.Stability Criterion

[1]Kinematic stability is achieved when a roller pair, which is momentarily disturbed from its equilibrium position by an external force, is capable of returning to a neutral position only because of corrective geometry changes. If the geometry of the contact between the two cylinders is such that the inequality that initiates this axial motion is changed to the equality of equation (27) by the axial motion, then the contacts will be kinematically stable. Therefore, to achieve kinematic stability, the object is to select the roller geometry that tends to satisfy equation (27) after the roller has been disturbed. The model chosen to evaluate the roller's tendency to return to equilibrium assumes the contacting surfaces at the roller ends to be both continuous and symmetric. Since these surfaces are rigidly connected, axial motion of one roller relative to the other will cause equal but opposite changes of contacting radius ratio at the roller ends. Therefore, the total stability question of the roller pair can be studied by considering the contact geometry changes at one end only. In plane **1** a stabilizing condition exists when

$$\frac{\partial}{\partial z}\left(\frac{R_{1b}}{R_{1a}}\right) < 0 \quad \ldots\ldots\ldots\ldots\ldots(29)$$

where z denotes a shift of cylinder a to the right as indicated in figure 12a. Since the contact geometry is symmetric, then in plane 2

$$\frac{\partial}{\partial z}\left(\frac{R_{1b}}{R_{1a}}\right) > 0 \quad \text{................(30)}$$

The sign of the inequality appearing in equation (29) is a consequence of the rollers tendency to move axially in the direction towards the end that possesses the smaller rolling radius. For the example selected in figure 12a(a), as roller a moves in the positive z direction relative to roller **b,** the radius ratio r_{lb}/r_{la} is required to decrease in value in order to restore equilibrium in accordance with equation (27). Considering the anti-symmetric behavior of the contact occurring in plane 2, just the opposite change would be required; that is, r_{2b}/r_{2a} is required to increase in value to restore equality between the radius ratios at each end. These stability requirements are reflected by equations (29) and (30) as shown. As previously mentioned, for roller bodies with symmetric ends, it is sufficient to consider the stability characteristics of the contacting geometry at only one end. In the analysis that follows, the contact occurring at the plane1 end or the end on the right side of the roller will be examined. The compliance of an axially disturbed roller pair to the criterion described in equation (29) will cause the rollers to eventually approach a condition of pure rolling where equation (27) is satisfied. If the axes are still skewed at this point, further motion will cause r_{2b}/r_{2a} to become larger than r_{lb}/r_{la}, which will tend to skew the axes in the opposite direction. This correction would tend to return the cylinders to their stable equilibrium point of pure rolling where these ratios are equal. In practice, a damped oscillation would most likely result. A roller pair exhibiting this behavior would be considered to have both axial as well as angular kinematic stability.

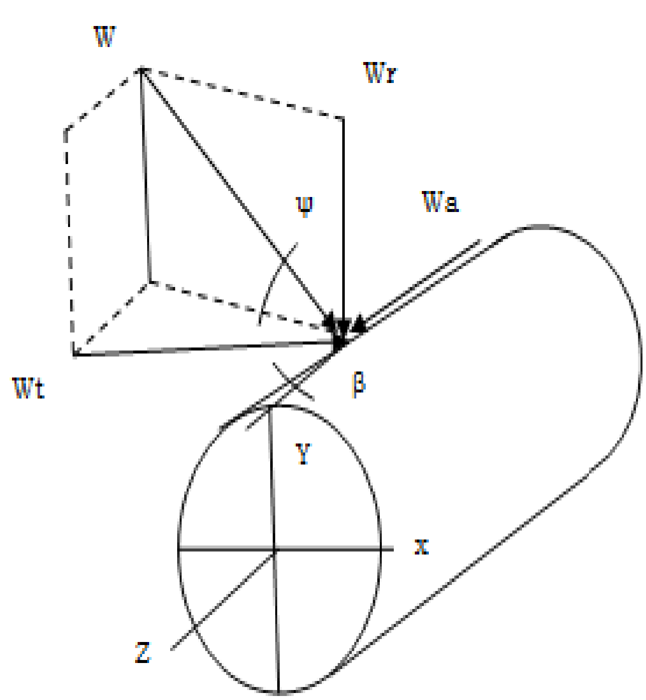
3.4.9.Mathematical analysis for ANSYS input

[15]The idea that axial displacement or deformation is supposed to show the possible thrust load on different kiln system components has made use of static analysis using ANSYS software. This analysis was accomplished for different values of rotational speed and skewing angle. This has brought different values for axial displacement or deformation of the kiln tire which shows that this effect will be reflected on the axial thrust load impact for the kiln components.

3.4.10. Axial thrust due to tire-roller contact surface

[15]Due to skewing of support roller the action of axial thrust load resembles that of the load acting on two meshing helical gears.

Figure 14. Loads acting on contact surface of support roller



$$W_r = W\sin\psi \tag{31}$$

$$W_t = W\cos\beta\cos\psi \tag{32}$$

$$W_a = W\cos\psi\sin\beta \tag{33}$$

W-Total Load

Wt-Tangential Load

Wr-Radial Load

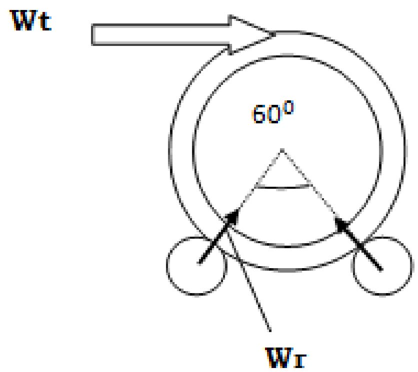
Wa-Axial Load

When support roller reaction analysis was statically done it was found that 63% of the total load of the kiln system has rested on the second basement roller pair. Since they are pair of rollers seated symmetrically at 60° to each other they equally share the load both in the x and y direction. And even the load in the x direction has been canceled due to opposite actions of the two support rollers.

Figure 15. Load Application areas on kiln tire where

Wt is tangential load applied due to drive motor and

Wr is reaction load due to the weight of kiln system



As the aim of this thesis is to show the effect of rotational speed on axial thrust of any rotary kiln for a fixed roller skew value, it has been tried to show with four different values of rotational speed.

Figure 16. Quantities of tangential load and torque obtained by varying rotational speed of kiln

Quantities to be determined	**RPM**	**1.75**	**2.0**	**2.1**	**2.3**
	Rad/sec	**0.18**	**0.21**	**0.22**	**0.24**
Force (KN)		48.16	65.55	71.94	85.62
Torque for a 2.14m arm (KN.m)		103	140.28	153.96	183.22

Figure16a. Quantities of tangential load obtained by varying skewing angle

Quantities to be determined	**Skewing angle (in degree)**	**0.21**	**0.32**	**0.42**	**0.57**
Force (KN) (Force in the z direction)		13.7	20.9	27.4	37.2

The defining input values for ANSYS software have been found to be load components in the x, y and z axes of the model. These loads were mathematically calculated using both static and dynamic analysis as

$F_y = W\sin\psi$ ………………………… (34)

$F_x = F_t = mw^2R$ ………………………… (35)

$F_z = w\cos\psi \sin\beta$ ………………………… (36)

Where

- W- total load acting on the pair of support rollers
- Ft- tangential load acting due to the torque given by the drive set to rotate the kiln
- m- mass of whole kiln system
- w- rotational speed of kiln
- R- arm radius of kiln shell
- Ψ- angle that the two support rollers subtend divided by two
- B- angle that support rollers are purposely skewed to generate axial displacement to balance the opposite thrust load

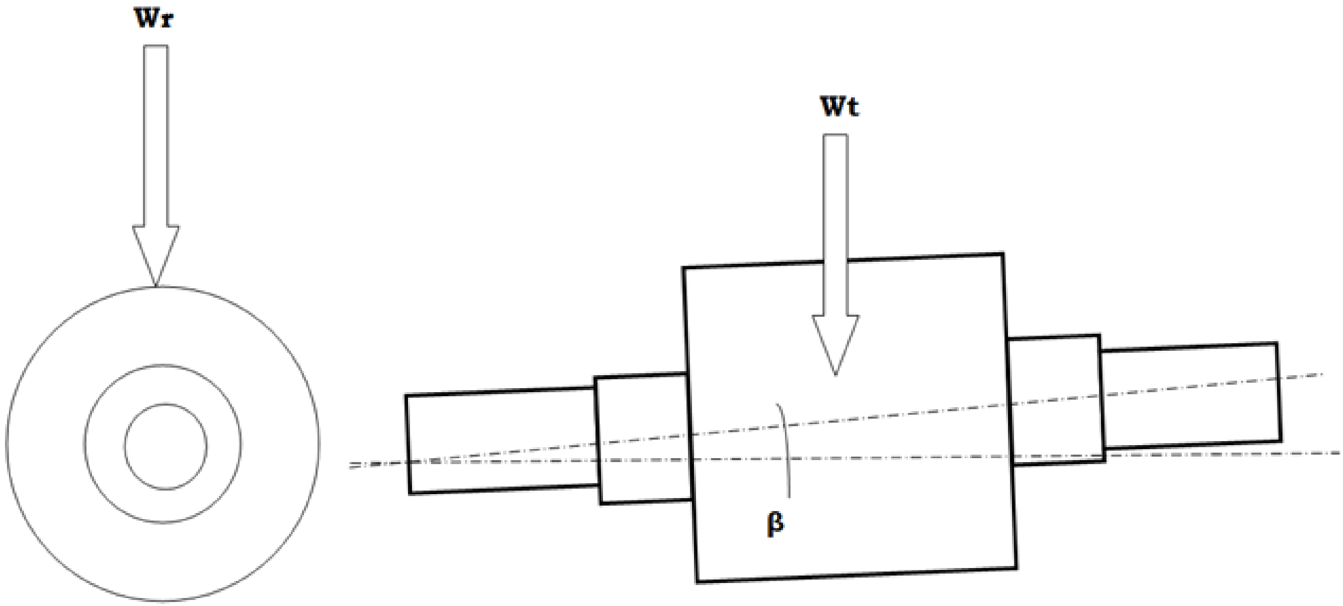
Figure 17. All force components for varying speed

Quantities to be determined	RPM	1.75	2.0	2.1	2.3
	Rad/sec	0.18	0.21	0.22	0.24
Force (KN)		48.16	65.55	71.94	85.62
Torque for a 2.14m arm (KN.m)		103	140.28	153.96	183.22
F_x(KN)		48.16	65.55	71.94	85.62
F_y(KN)		2158.2	2158.2	2158.2	2158.2
F_z(KN)		37.2	37.2	37.2	37.2

Figure 17a. All force components for varying skewing angle

Quantities to be determined	Skewing angle (in degree)	0.21	0.32	0.42	0.57
F_x(KN)		85.62	85.62	85.62	85.62
F_y(KN)		2158.2	2158.2	2158.2	2158.2
F_z(KN)		13.7	20.9	27.4	37.2

Figure 18. Areas of Tangential load application from top view and radial load from side view

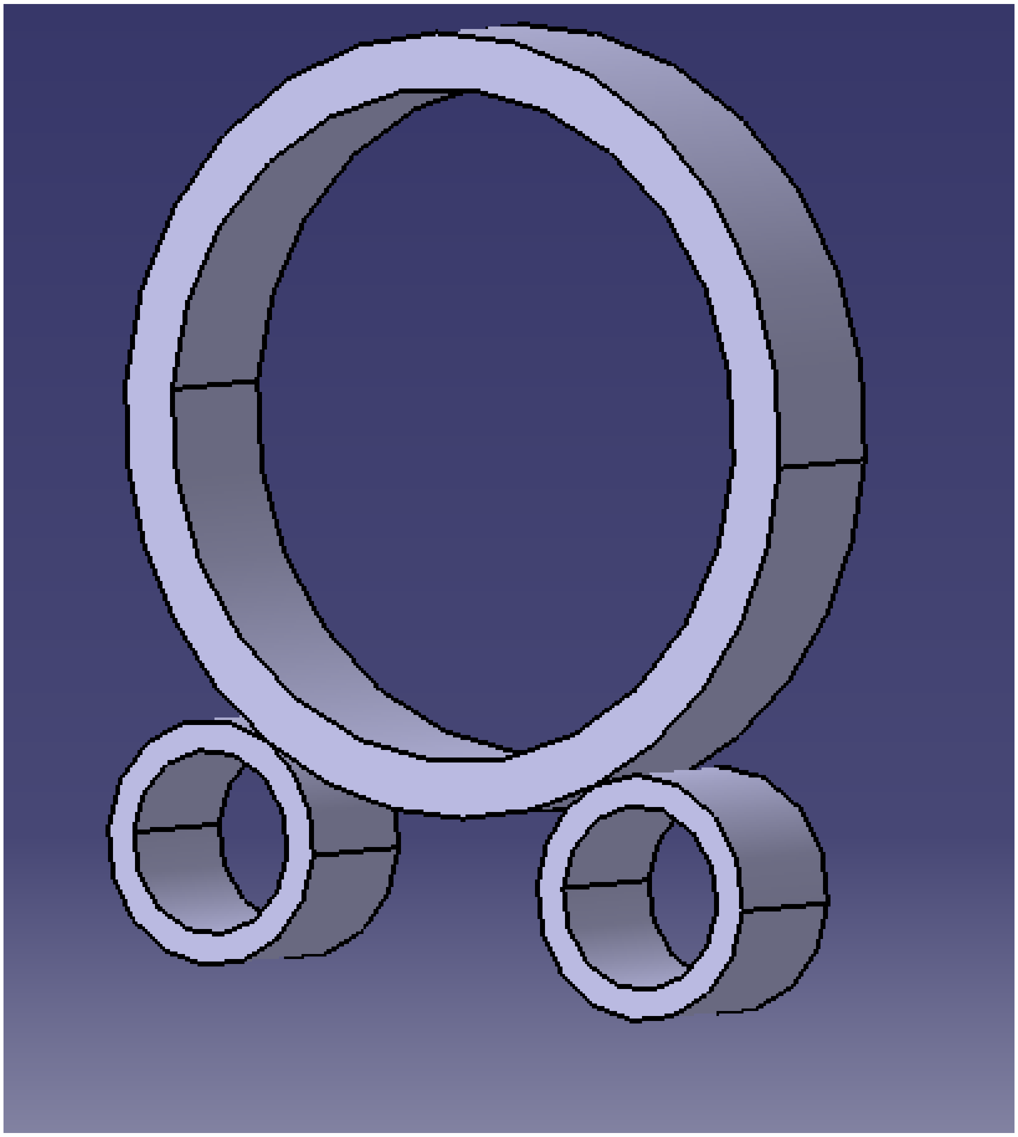


3.5. CONDITION

This chapter particularly deals with the question that when this thesis is conducted 'how the desired output could be obtained.' The whole research is all about treating the two variables by describing using different modeling methods. Since the thesis considers the effect of rotational speed and skewing angle on the axial thrust of a rotary kiln we need to put some conditions while trying to show the effect of speed and skewing angle though there are other factors that determine the magnitude of the axial thrust. Axial movement of rotary kiln due to skewing of rollers will not create huge wear damage on the surfaces of support rollers and tire rings unless there faces a resistance to the motion due to skewing. If we are able to control the motion by skewing the support rollers in both directions from the absolute parallel state of axes of support rollers and rotary kiln we can keep the axial thrust less harm to components of kiln system. Here the principle of kinematics and kinetics is applied. Controlling the axial motion with kinetic correction will produce considerable thrust on components. Usually the axial thrust of rotary kiln nowadays is controlled by hydraulic thrust roller unit. However, this is totally kinetic or force correction which will create significantly large thrust force that has to be withheld by disc of the thrust unit and this will accelerate shortening the life of thrust roller.

- It is assumed that the varying rotational speeds will be taken for every maximum value of other parameters
- It is also assumed that the varying skewing angles will be taken for every maximum value of other parameters
- Even though the slope of base plates, support rollers and tire rings are susceptible to change when the kiln is in operation for longer time for the sake of my analysis it is considered to be the design value which is 3%
- The surface conditions of all contacts between the tire rings and support rollers are assumed to be uniform and taken from the design values.

Figure 21 CATIA modeling for tire and roller assembly



Nearly all components which exist in real situation of rotary kiln system have been modeled using CATIA V5. And all components were assembled based on the order of instruction manual and the assembly is shown in figure 19. However, capacity of the personal computer which has been engaged to execute all the tasks and efficiency of ANSYS software cannot allow for analyzing the real motion of the process.

CHAPTER FOUR

4. RESULT AND DISCUSSION

4.1. RESULT

The ANSYS result shows directional deformation in Z-axis has been changed as a result of change in tangential force that would cause the kiln to rotate. And directional deformation in the Z-axis has also been changed as a result of change in the skewing angle. The tangential force was made to change by changing rotational speed of kiln. The directional deformation for different number of rotation has shown an increment.

Results of equivalent strain and stress were also shown for resultant load fed to the software. These quantities were also shown changes as a result of change in number of rotation of kiln instead of speed.

The three quantities namely directional deformation, equivalent strain and equivalent stress were obtained for different tangential force and number of rotation. The result shows for all quantities that there is no change for tangential force but considerable difference was observed by varying number of rotation.

Figure 21a Meshing results of ANSYS

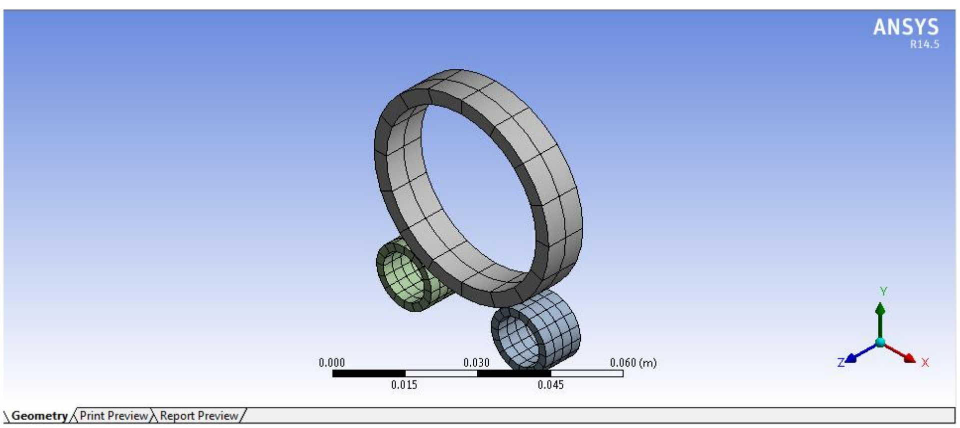
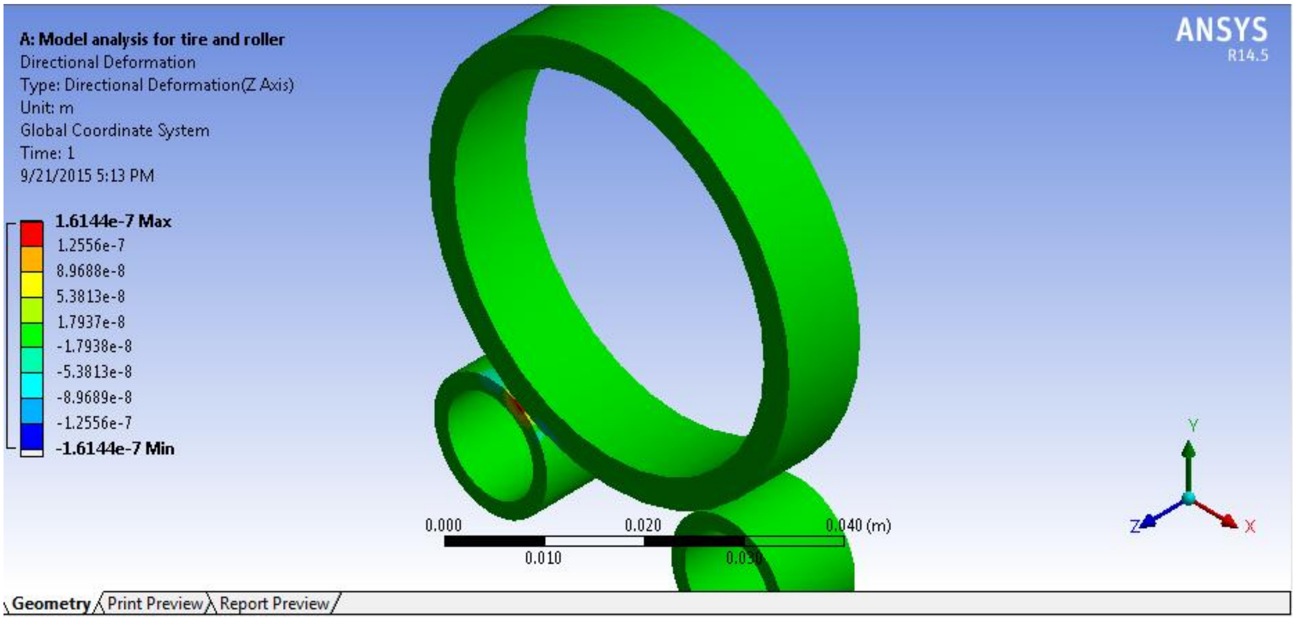
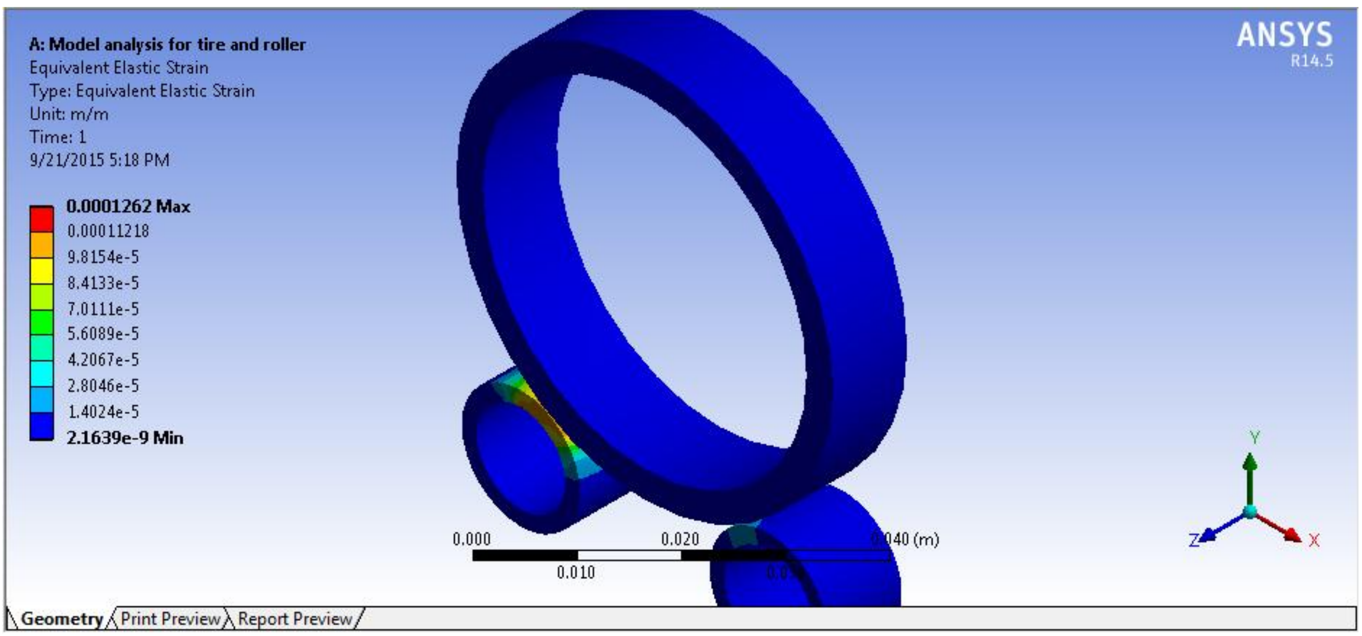


Figure 22 ANSYS Results for one revolution, 360°





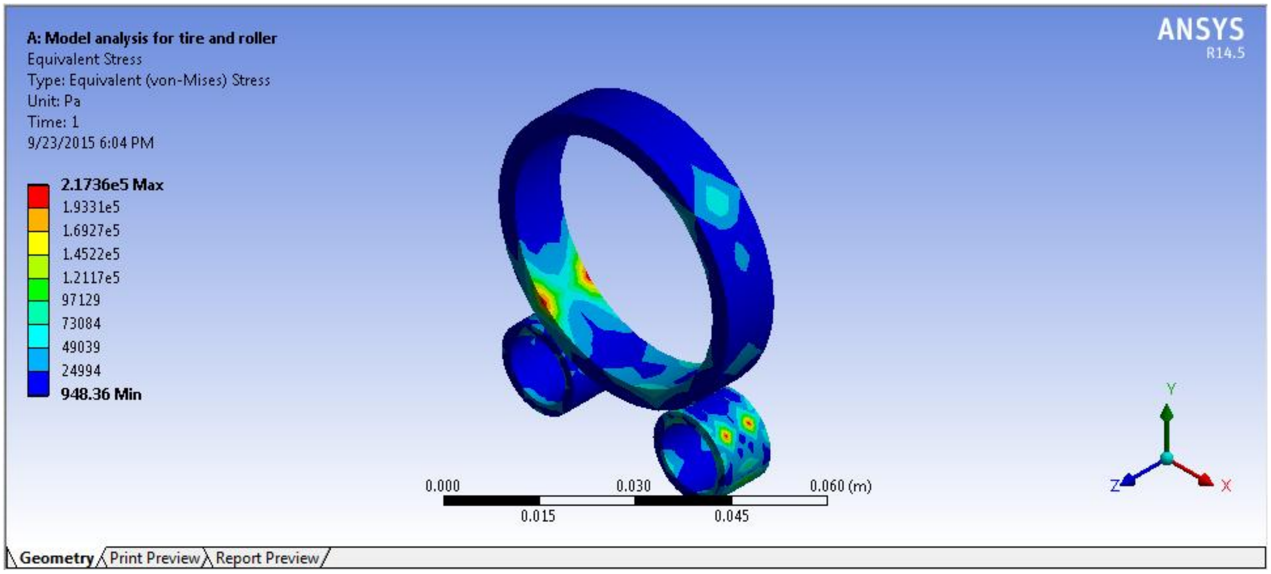
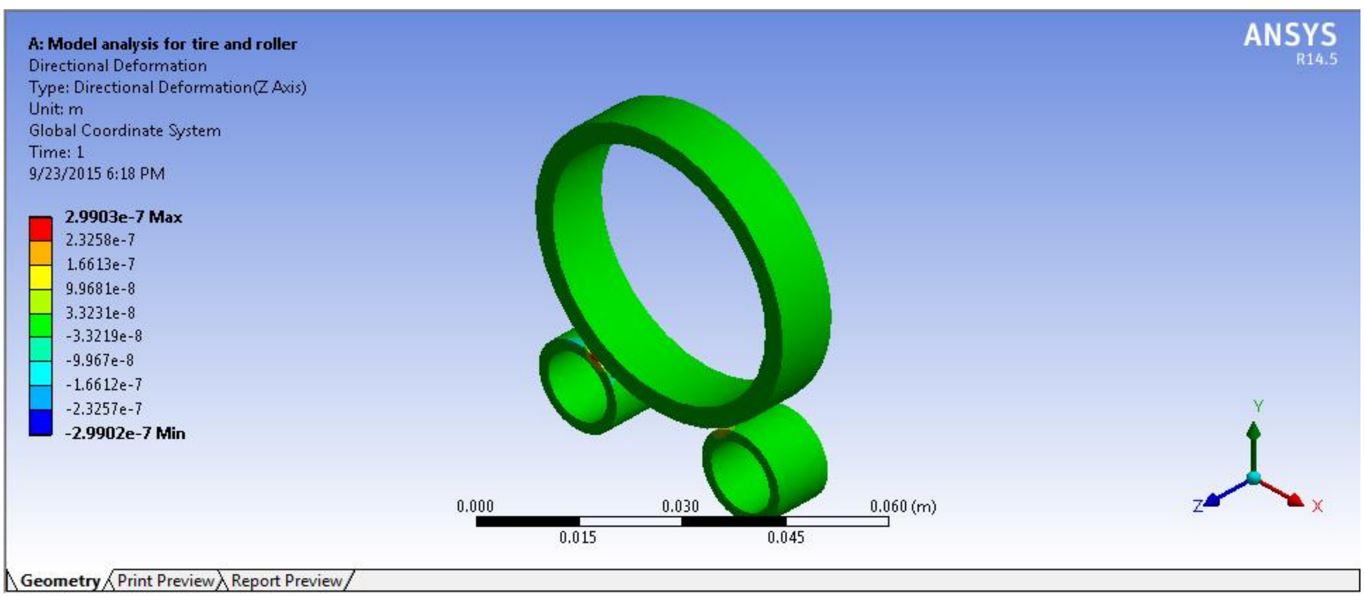


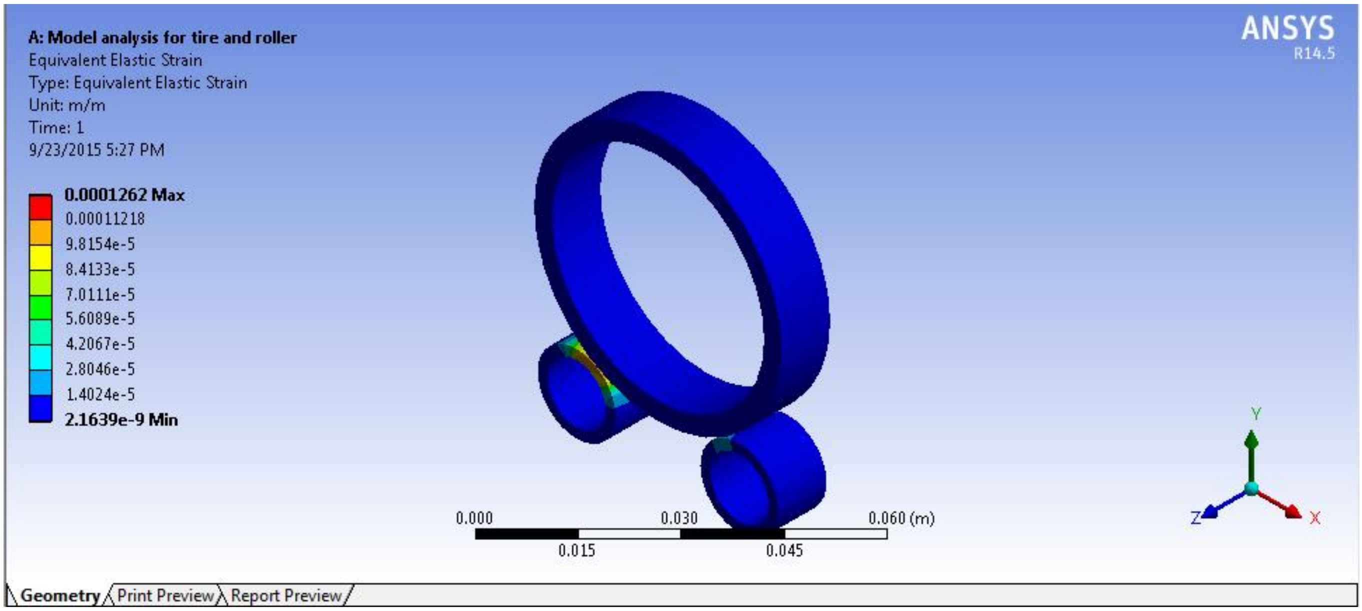
Figure 23 Tabulated results of ANSYS for one revolution

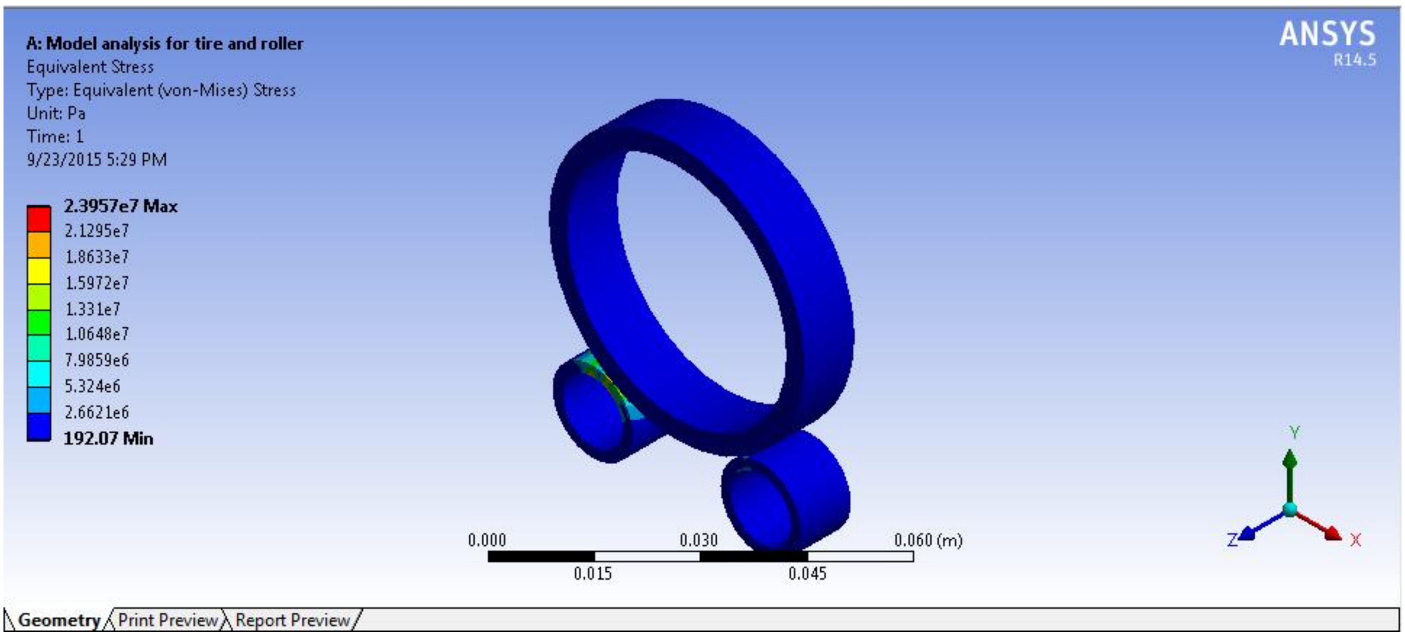
Object Name	*Directional Deformation*	*Total Deformation*	*Equivalent Elastic Strain*	*Equivalent Stress*
State	Solved			
Scope				
Scoping Method	Geometry Selection			
Geometry	All Bodies	3 Bodies	All Bodies	
Definition				
Type	Directional Deformation	Total Deformation	Equivalent Elastic Strain	Equivalent (von-Mises) Stress
Orientation	Z Axis			
By	Time			
Display Time	Last			
Coordinate System	Global Coordinate System			
Calculate Time History	Yes			
Identifier				
Suppressed	No			
Results				
Minimum	-1.6144e-007 m	3.3142e-017 m	2.1639e-009 m/m	192.07 Pa
Maximum	1.6144e-007 m	1.3265e-002 m	1.262e-004 m/m	2.3957e+007 Pa

Minimum Occurs On	Roller2	Tire	Roller1	
Maximum Occurs On	Roller2			
Minimum Value Over Time				
Minimum	-3.2999e-006 m	3.3142e-017 m	2.5041e-011 m/m	1.7551 Pa
Maximum	-6.8009e-011 m	9.3683e-003 m	2.888e-008 m/m	4661.1 Pa
Maximum Value Over Time				
Minimum	1.1021e-010 m	3.9215e-003 m	5.0841e-008 m/m	9335.5 Pa
Maximum	3.2999e-006 m	5.0991e-002 m	3.8723e-003 m/m	7.4718e+008 Pa
Information				
Time	1. s			
Load Step	1			
Substep	54			
Iteration Number	246			
Integration Point Results				
Display Option			Averaged	

Figure 24 ANSYS Results for two revolutions, 720°

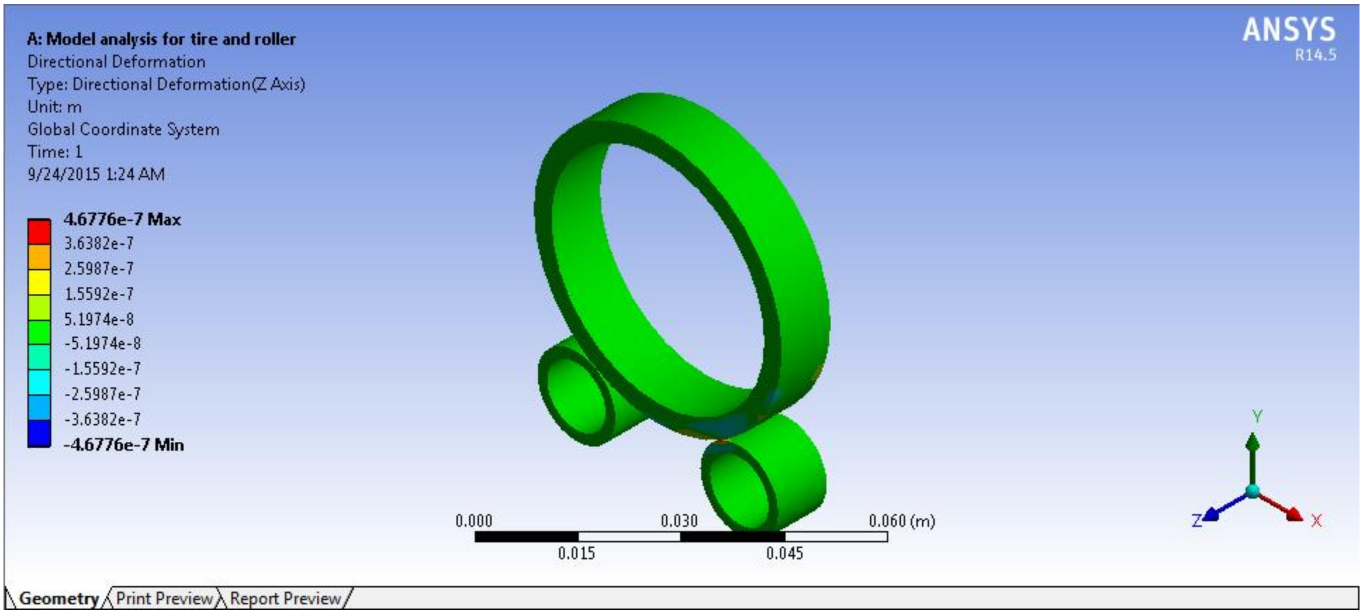


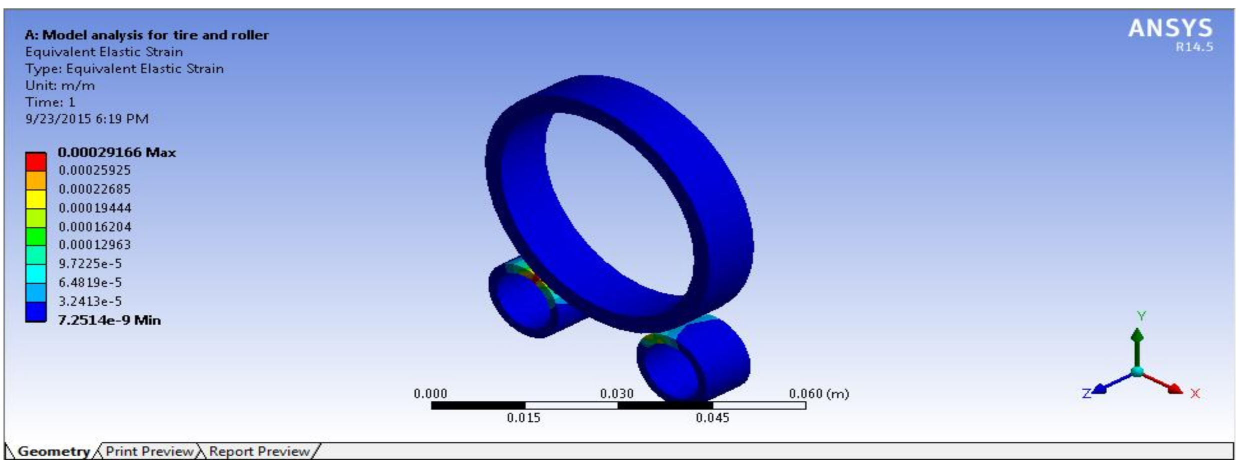




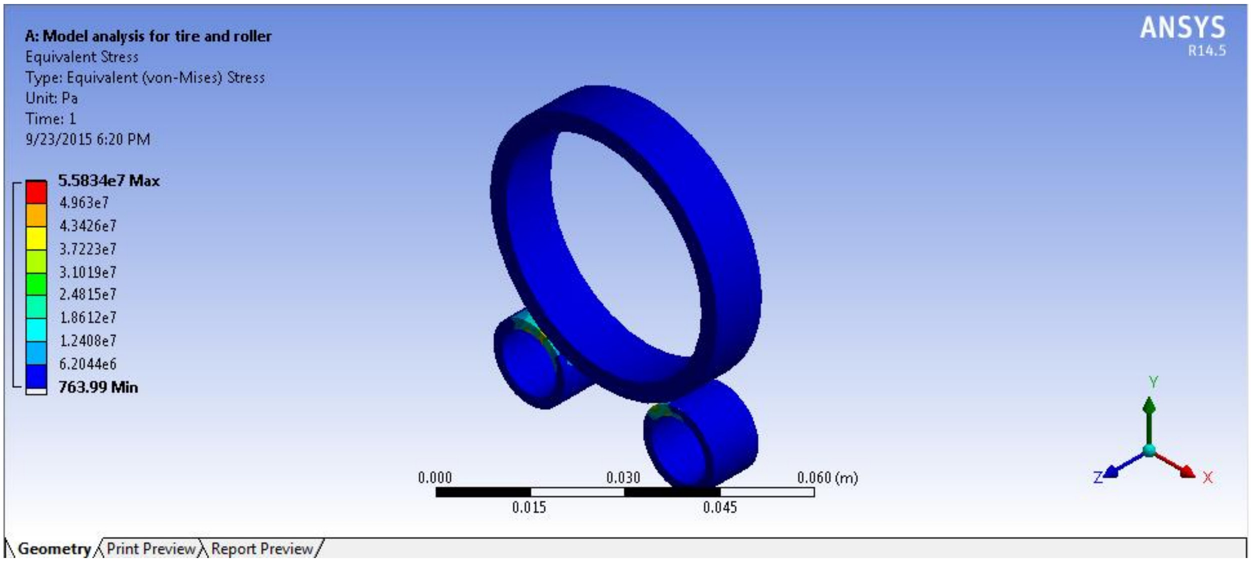
Dimensions to be determined	Rotational speed of kiln(RPM)			
	1.75	2.0	2.1	2.3
Axial Displacement for one revolution (in meter)	2.9903×10^{-7}	2.9903×10^{-7}	2.9903×10^{-7}	2.9903×10^{-7}
Equivalent strain	1.262×10^{-4}	1.262×10^{-4}	1.262×10^{-4}	1.262×10^{-4}
Equivalent stress(Pa)	2.3957×10^{7}	2.3957×10^{7}	2.3957×10^{7}	2.3957×10^{7}

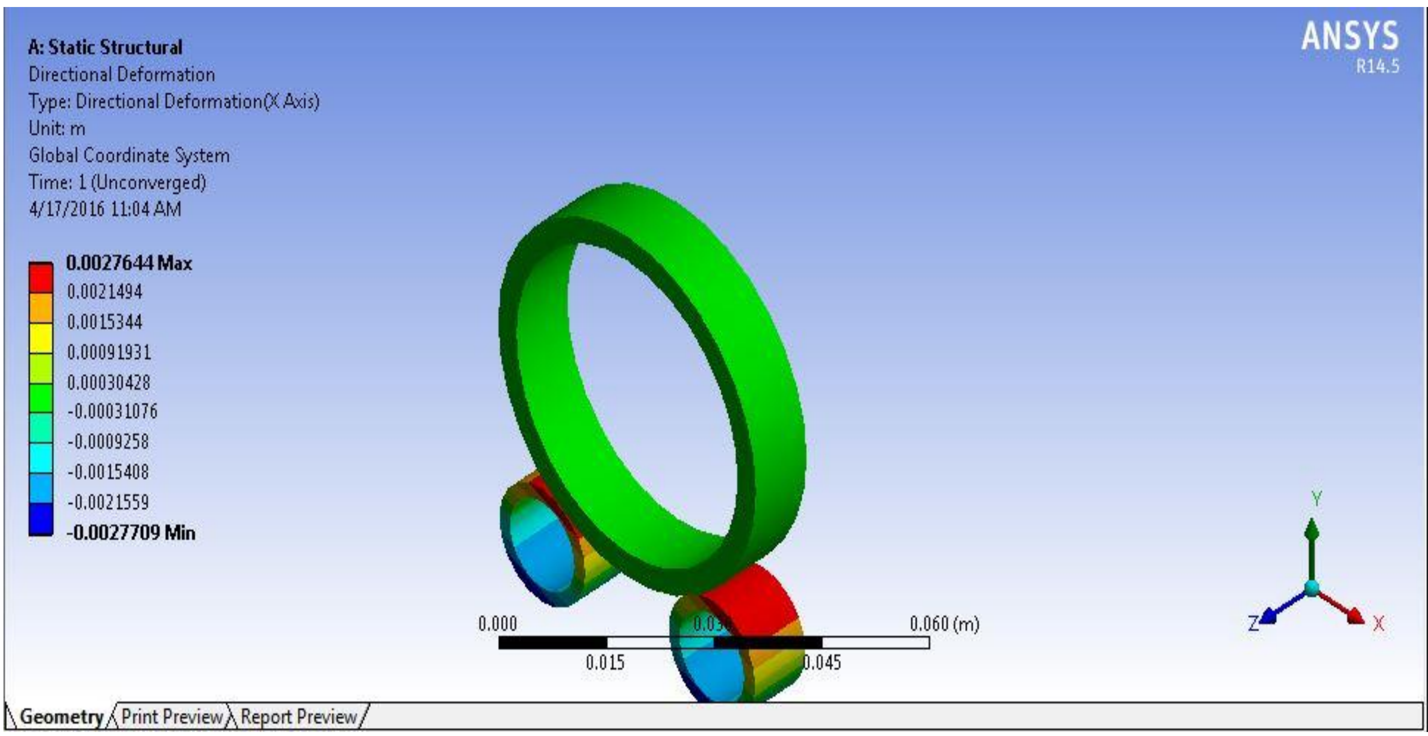
Figure 25 ANSYS Results for three revolutions, 1080°





Dimensions to be determined	Rotational speed of kiln(RPM)			
	1.75	2.0	2.1	2.3
Axial Displacement for one revolution (in meter)	4.6776×10^{-7}	4.6776×10^{-7}	4.6776×10^{-7}	4.6776×10^{-7}
Equivalent strain	2.9166×10^{-4}	2.9166×10^{-4}	2.9166×10^{-4}	2.9166×10^{-4}
Equivalent stress(Pa)	5.5834×10^{7}	5.5834×10^{7}	5.5834×10^{7}	5.5834×10^{7}





4.2. DISCUSSION

Initially the static analysis was chosen so as to see the change in directional deformation with respect to tangential force because the force is subject to change due to change of rotational speed. However, results have shown that the skewing angle change has more direct impact on axial load than the speed change particularly for low speed-equipment. However, number of revolutions accumulates axial displacement for very long operation time. Much more significantly, it has been observed that change in skewing of support rollers from 0.4° to 0.57° produced an additional load roughly 34% of the original one. Since mass of rotary kiln system is extremely huge the speed with which it rotates is very small to bring a considerable value to make change in the value of tangential force. In real situation as of Mugher Cement Factory for kiln with favorable working condition it has a record of running for 11 months without stoppage. Records have shown that during such time an axial displacement of roughly 5centimeter was observed for a perfectly aligned support rollers. However, a skewing 11mm push of support roller bearing to the opposite has brought the kiln back to its normal position in two months. In this actual work result, one can observe that in 60days 198720revolutions were made for a full capacity running which is 2.3rpm. With this number of revolution the kiln was back from 50mm away to its normal position. In every revolution an axial displacement of $2.516*10^{-4}$ mm has been made. However, ANSYS result has shown tire displaced $1.6144*10^{-4}$ mm for one revolution. This is a 35.8% deviation from the actual result.

CHAPTER FIVE

5. CONCLUSION, RECOMMENDATION AND FUTURE WORK

5.1. CONCLUSION

At the beginning, the idea of this thesis is to show the effect of rotational speed and skewing angle on axial thrust of a rotary kiln. The result of both mathematical and software analysis have shown that effect of change in skewing angle of support rollers have larger than that of rotational speed on axial thrust of rotary kiln for the fact that (unlike equipment such as pumps, helical gearboxes, fan etc.. which have extremely high speed as compared to kiln) speed of this equipment is so small that the result of change in tangential force will considerably be insufficient to bring about axial thrust due to skewing. However, the result has also shown that the number of revolution the kiln makes during its operation has a significant effect to axially displace depending on the sense of skewing direction. Most importantly skewing angle changes have a considerable change in axial thrust.

5.2. RECOMMENDATION

In cement or mining industry as a whole axial thrust of any rotary kiln causes a massive destruction on different kiln system components as it has been observed from different experience and exposure. Having been in cement factory for nearly ten years as mechanical engineer, I can witness that the damage has been causing potential loss in terms of increasing cost of production for replacing damaged components or loss due to stoppage for maintenance. Hence I herewith recommend the following procedural steps for short and long term solution to minimize damage of components and ultimately eliminate unnecessary axial thrust of rotary kiln:

- It is advisable to control axial thrust using kinematic methods instead of force control or kinetic action.
- Although a kinetic or force correction, in the form of a roller flange or bumper, is sometimes required to ensure that gross axial motion cannot occur, the kinematic correction is unquestionably more desirable.

- The kinematic correction, which results in self-centering action, can be implemented without seriously affecting either the mechanical efficiency or durability of the contact.
- Axial motion that is not corrected kinematically will create significantly large thrust forces that have to be withheld by flanges or thrust bearings
- As effect of rotational speed has been studied as one factor to affect axial thrust it is also important and compulsory to complete study of the other factors to apply in formulating a controlling setup that could be developed into a technology which would make use of flexible skewing of support rollers that acts depending on the magnitude of thrust exerted on kiln system components to relieve pick thrust situations.

5.3. FUTURE WORK

This thesis, as tried to note in the introduction part, is to investigate the effect of rotational speed and skewing angle on axial thrust of rotary kiln and it is done. However, the effect of these two changes and other factors should be further analyzed for technology application. Then it will have a potential attraction for sponsors to invest to change it to technology.

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