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ACHIEVING MECHANICAL STABILTY OF ROTARY KILN BY FEM

Sumesh Krishnan

B.E, M.B.A., M.Tech

ABSTRACT

Rotary kilns are found in many processes that involve solids processing. These include drying, incineration, mixing, heating, cooling, humidification, calcinations ,reducing, sintering and gas-solid reactions (Jauhari et. al. 1998). The most common and industrially important application of rotary kilns is in cement production; all major producers use the rotary kiln as their equipment of choice.

Rotary kilns are amongst the most well-established unit operations in the process industry, yet are amongst the least understood. They can be used for 3 purposes: heating, reacting and drying of solid material, and in many cases, they are used to achieve a combination of these aims. In the design of kilns, there are four important aspects to consider from a process engineering point of view, and these are heat transfer, flow of material through the rotary kiln, gas-solid mass transfer and reaction.

Roller and shaft of Cement Rotary Kiln supporting roller is exposed to high stresses under heavy working situation. High Stresses because of big kiln dimensions cause damage to roller and shaft of supporting roller mechanisms. This requires that, decreased stresses provides longer life and decreased damaged to roller and shaft of supporting roller with material properties and working conditions..

Active researching and development of the mechanics of cement factories equipment's are not studied in the cement industry. Researching roller and shaft of supporting roller will be useful in the cement industry and is not a good candidate for experimentation due to economical reasons. However, engineers use model in the computer analysis program with original dimensions of subjects and engineers do complex model analysis with the aid of computer. The material properties and working situations are given to analysis program on the computer veritably. Roller and shaft model is drawn and analysis done in order to get critical stresses results.

I. INTRODUCTION

The investigations show that the causes of refractory failure and poor durability of refractory in large diameter kilns are due to the following factors.

- Grade & lining methods of refractory
- Operational influence
- Mechanical stresses on the refractory.

The paper also discusses the causes of refractory lining failure, component breakdown with reference to mechanical stresses which are due to the following factors:

- Kiln shell ovality

- Kiln shell warping
- Kiln alignment
- Mechanical condition of kiln

II. KILN ALIGNMENT

2.1 Definition

A kiln is supposed to be aligned if its rotating axis is rectilinear. In other words, if the rotating centers of shells at centre of tyres are joined together, these should form a straight line, in horizontal as well as vertical plane.

2.2 Cold Kiln Alignment

This can be carried out by taking the measurements while the kiln is stopped. The cold alignment has many limitations, however, a few are listed below:

- measurements are not very accurate
- wear of tyres, rollers and chairs is not considered
- effect of temperature is not considered

Despite above limitations, this method of alignment is in use in many cement plants, since its cost is quite low as compared to the hot kiln alignment.

2.3Hot Kiln Alignment

In this method, the readings are taken while the kiln is operating at full production. The readings obtained are fairly accurate. However, this method of alignment is costly.

2.4 Alignment Requirement

For improving the refractory life the kiln should be aligned every 2 years preferably by hot alignment method.

III. MECHANICAL CONDITION OF KILN

3.1 Kiln Shell Runout

3.1.1 Radial and Facial Runout of Girth Gear

To obtain extended life of girth gear and pinion the runouts should not exceed the acceptable limits. When the readings show excessive runouts the girth gear should be re-aligned.

3.2 Pinion-Girth Gear mesh

3.2.1 Skew of Rollers

Excessive skewing and wrong skewing of rollers may create heavy thrust reaction on the bearings and at times vibrations which are detrimental to the lining life.

3.3 Lubrication of Tyre

Life of refractory lining can be improved by:

- Maintaining shell ovality within closely set limits by correcting tyre clearance

- Continuously monitoring the tyre creep, temperature of tyre and shell. Cooling the shell when the limits for creep and temperature are critical.
- Avoiding rapid heating and cooling of kiln
- Checking and correcting alignment of kiln every 2 years and
- Maintaining the kiln in a good mechanical condition

- extended lifetime of the refractory lining and elimination of the lining loss caused by the rotary kiln mechanical instability
- energy savings
- enhanced lifetime of the shell, tyres, stop-blocks, support rollers, shafts and bearings
- reduced wear of a gear ring, pinion and drive
- elimination of cracks and overheating of the kiln shell

Typical Kiln Failure Symptoms

Hot bearings		**Cracks**		**Mechanical wear**		**Lining loss**	
On bearing journal	On thrust journal	In kiln shell	On roller shaft	On support roller and tyres	Between tyre and kiln shell	Between tyre	Under tyres
Kiln geometry		Kiln geometry		Kiln geometry		Kiln geometry	
		Kiln shell ovality		Kiln shell ovality		Kiln shell ovality	
Kiln crank		Kiln crank Kiln crank				Kiln crank	
Axial balance				Axial balance			
Lubrication		Corrosion				Lining quality	

An extensive problem if we include thermal radiation (loads), chemical mixing, particulogy would be numerically expensive to solve. Optimal state of balance, dynamics, power savings, reduced maintenance, increased productivity can only be achieved in near future when cost of computation is more economically feasible

3.4 Kiln Geometry Analysis

The high reliability and the ability to predict breakdowns of a rotary kiln critically depend on its geometry. Changes such as sinking of foundations, uneven mechanical wear or an inadequate repair can lead to overloading of individual components. This condition then gradually develops into a full-fledged damage and production loss.

Rotary kilns should be therefore carefully aligned in order to reduce to minimum torsion and stress in the kiln shell, i.e. the factors cause excessive mechanical wear and can negatively influence the lifetime of refractory lining, rollers, bearings and other components.

Analyzed elements:

- kiln axis (spatial course)
- kiln slope
- roller operating angles
- deviation of supporting roller axes from the kiln axis (spatial)
- diameters and profiles of the supporting rollers
- diameters and profiles of the tyres
- tyre and gear ring wobbling
- thermal differences

IV. MATERIALS AND METHODS

4.1 Finite Element Analysis

In this finite element analysis the continuum is divided into a finite numbers of elements, having finite dimensions and reducing the continuum having infinite degrees of freedom to finite degrees of unknowns. It is assumed that the elements are connected only at the nodal points. The accuracy of solution increases with the number of elements taken. However, more number of elements will result in increased computer cost. Hence optimum number of divisions should be taken. In the element method the problem is formulated in two stages

The Element Formulation

The System Formulation

4.2 Basic Steps in the Finite Element Method

- Discretisation of the domain
- Incorporation of boundary conditions
- Formation of the element loading matrix
- Formation of the overall loading matrix
- Solution of simultaneous equations
- Calculation of stresses or stress resultants

V. RESULTS

In the scope of:

- FEA analysis of the support roller;

General aim of analysis:

- determination of maximal stresses values and location in roller's shaft material;
- guidelines for improvements and reinforcement methods to reduce stresses;

5.1 Actual Stress Test Results

5.1.1 Brief

1. The analysis was performed according to research requirements.
2. the purpose / reason of the analysis is given subsequently.
3. the document shows the condition of the facility as on: 2013-11-28 (reception of the latest data);
4. all data included herein and in the attached appendices were stored in the computer database;
5. particularly important remark/ guideline – **in bold** for easier identification,

(R) - remark/ guideline – repetition/ reminder of the contents of the previous inspection report which is still valid,

- additional information/ theoretical foundation for easier understanding of the remarks/ guidelines.

5.1.2 Description of the problem

The plant is reporting problems of abnormal sound coming form the support roller of rotary kiln #1 - first support pier from the inlet side. In the past there was a breakdown of the roller shaft. Based on the email correspondence and revision of documentation, it has been stated, that there is a need of FEA analysis of roller shaft to determine the stresses value for the given (actual) load parameters.

Data and assumptions for analysis

Data received from the industry

- construction drawing of the roller – according to the Drawing 1 – see enclosure no.1;
- construction drawing of roller raceway - according to the Drawing 2 – see enclosure no.2;
- construction drawing of roller shaft – according to the Drawing 3 – see enclosure no.3;
- roller shaft material EN9 with technical specification according to the enclosure 4 – BS070M55(EN9), R_m = 715MPa – ultimate tensile strength;
- total weight of the object (including charge material) considered as the load to the support system m_t = 416tons;
- the most loaded support we consider middle pier (P2), force acting on the rollers equals 40% of total weight of the object R = 166.4tons;
- angle of support rollers at the kiln center is 60^0, load component acting on the single roller F_R = F_L = 96.1tons; Load per roller = (1273.18/2)*cos 30
- coefficient of dynamic load (considering 3.3 RPM) we assume 1.1 results in dynamic load value of **FR(=L) dyn = 105.7tons (per single roller)** .
- for the complementary analysis there should be also included tension coming from the fitting of the raceway on the shaft. It is difficult to determine the pressure of each roller which depends on the manufacturing and

assembly quality. We have not taken this into consideration but the client should notice that this will additionally increase stresses value inside the shaft;

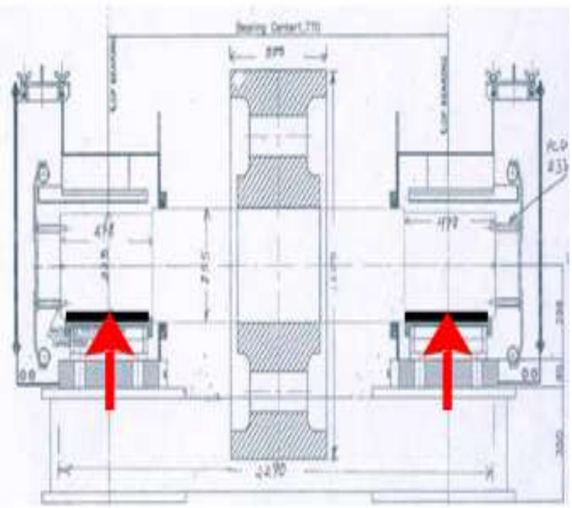
- we have assumed that the contact between roller and tire raceway is continuous on the total available width of ring raceway, coefficient of dynamic load is considering some additional increases of the load force but in typical, proper working conditions,
- we have assumed there is no hidden crank of shell and no large misalignment of support system axis.

5.1.3 FEA Analysis and Results

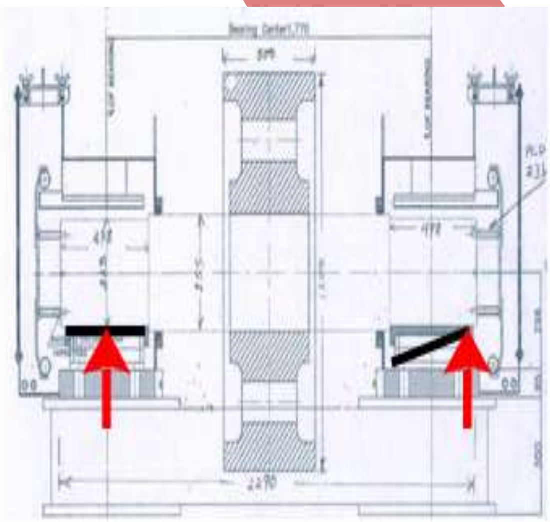
Drawing below is showing the construction of the roller assembly.

For our analysis we have considered different cases of bushings position.

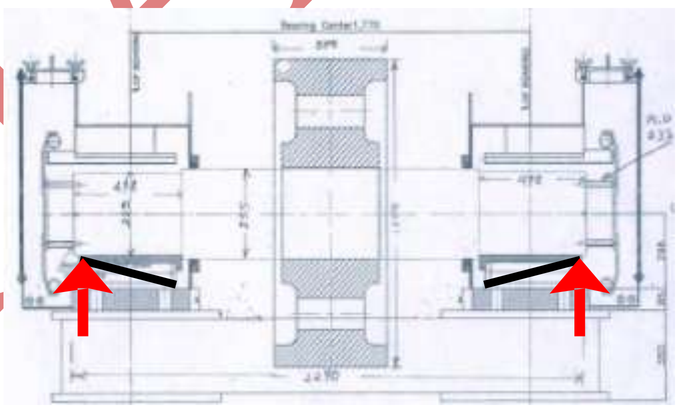
If the bearing is not self aligning the following cases can appear:



Case 1. Bushings Parallel to the Shaft Journal



Case 2. Right Bushing Nonparallel to the Shaft Journal

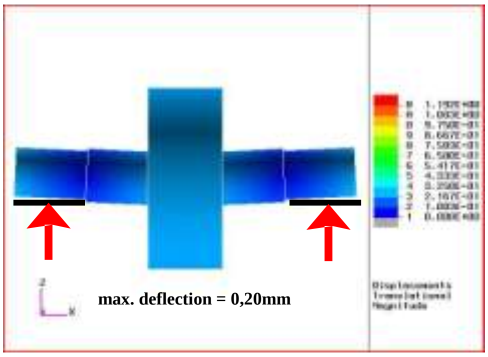


Case 3. Right and Left Bushing Nonparallel to the Shaft Journal

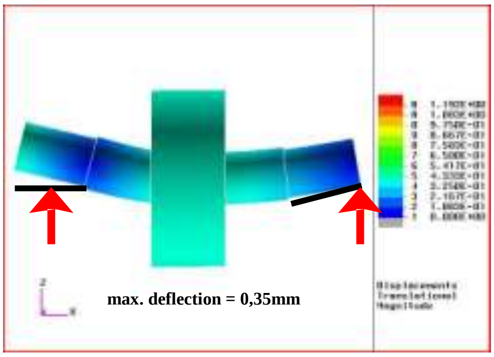
In each case the distance between reaction forces position is different.

In each case the equivalent of deformation and stress distribution is different also (see next pages).

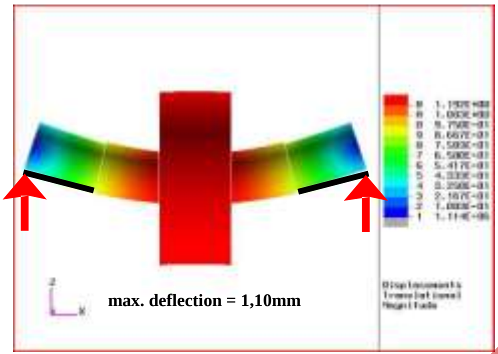
Finally, if the bearing is not self aligning, the following cases of deformation can appear:



Case 1. Bushings parallel to the shaft journal.



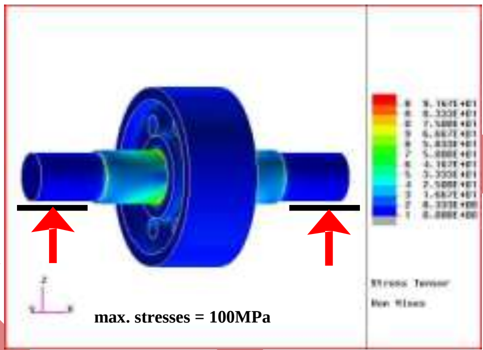
Case 2. Right bushing nonparallel to the shaft journal.



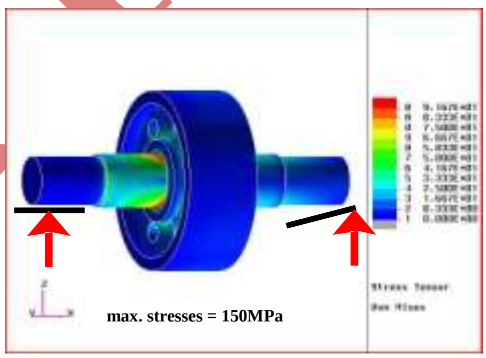
Case 3. Right and left bushing nonparallel to the shaft journal.

5.1.4 Deformation of the Roller Shaft, For Non-Self Aligning Bearings

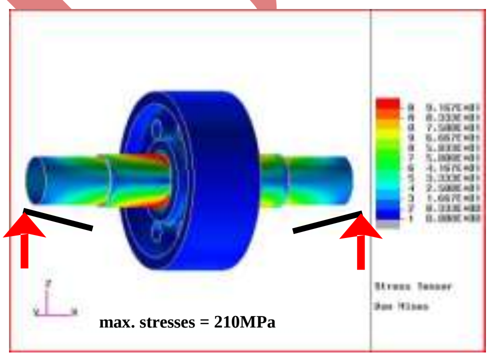
And finally, if the bearing is not self aligning, the following cases of stresses can appear (calculated according to Huber hypothesis):



Case 1. Bushings parallel to the shaft journal.



Case 2. Right bushing nonparallel to the shaft journal.



Case 3. Right and left bushing nonparallel to the shaft journal.

Stresses on the Roller Shaft (Huber Hypothesis)

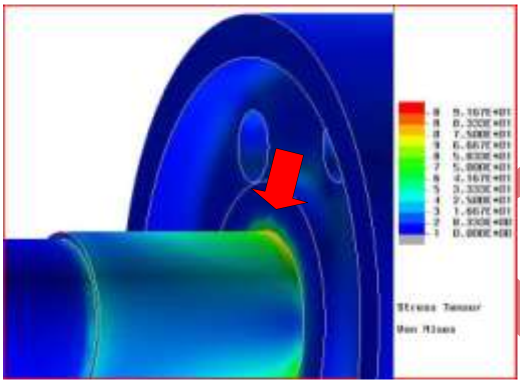
Strength Results Determined For Each Case.

Non Self Aligning Bearings

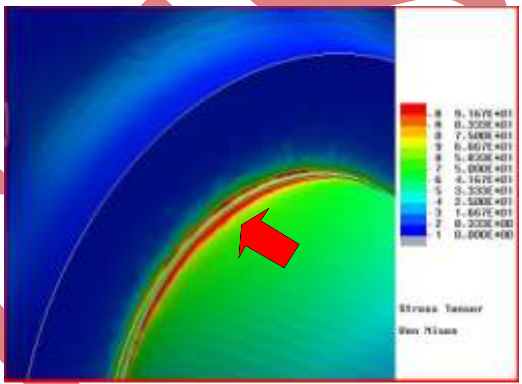
Case Number	Maximal tensile stress	Maximal shaft deflection
1	100 MPa	0.20mm
2	150 MPa	0.35mm
3	210 MPa	1.10mm

5.1.5 Commentary to the Obtained Results

Stop The most critical point of the shaft is always an entrance of the shaft inside the raceway bushing in in the analyzed cases the effect is amplified by the small step on the shaft 55mm wide 360 mm of diameter. This is the place of stress accumulation what is shown on the picture bellow.



Stresses on shaft and raceway joint (case 1)



Stresses on the small step of the shaft (case 1)

Fig Stresses On Shaft &Raceway Jont **Fig.5.6Stresses On Small Step Of The Shaft**

Stresses obtained during analysis should be compared to the once resulting from the shaft material strength point of view. For the shaft material ultimate tensile strength value (R_m=715MPa; value taken from the material specification) we can calculate material effort for cyclically changing load (symmetrical compression/ tension)

should not exceed k_{rc}=118MPa (krc=0.5kr=0.5(0.33R_m)).

From the table you may see that only in case of perfect alignment of the roller shaft and sleeve bearing this condition is assured. But such perfect conditions are not easy to assure. It could be a crank of shell or not properly installed shim under the roller housing and we are suddenly over the stress limit.

As far as we concerned, such construction (length and diameters) of roller's shaft is improper in reference to the load condition, even if there is no additional forces acting on the kiln. 100MPa (case 1) vice versa 118MPa (limit for used material) is too small reserve and doesn't assure not limited live of roller from fatigue point of view.

Guidelines

In relation to the above we suggest the following:

In urgent mode

1. **STOP** check and assure proper alignment of the bearing sleeve to the shaft journal. Our simulation of different cases (1, 2 & 3) shows that slope of the bearing sleeve should be exactly like the global roller shaft

slope. Any deflection of the frame under the roller which will not be compensated by the proper shimming will created appearance of overload (even up to 100% more!);

In preventive mode

2. execute kiln alignment to optimize load to each roller;

3. check hidden crank (dynamic roller's deflection);

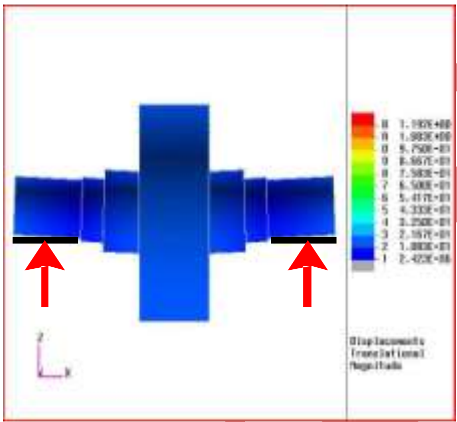
As a final aim

4. measure directly the real value of load per tire to confirm calculation data (the simulation of weights). If there are any differences assume proportional change of calculated stresses and deflections;

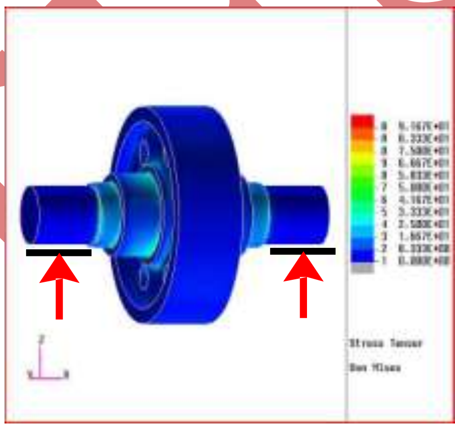
<u>Stop </u>modify construction of the roller installing the busing on the shaft – inside the raceway (1000mm of length, OD=455 mm, ID=355mm; it requires boring of raceway and assure proper small interference fit).

Tensile stress value in this case will go down to 50MPa-50% lower (for case 1).

See below graphs with results of appropriate simulation;

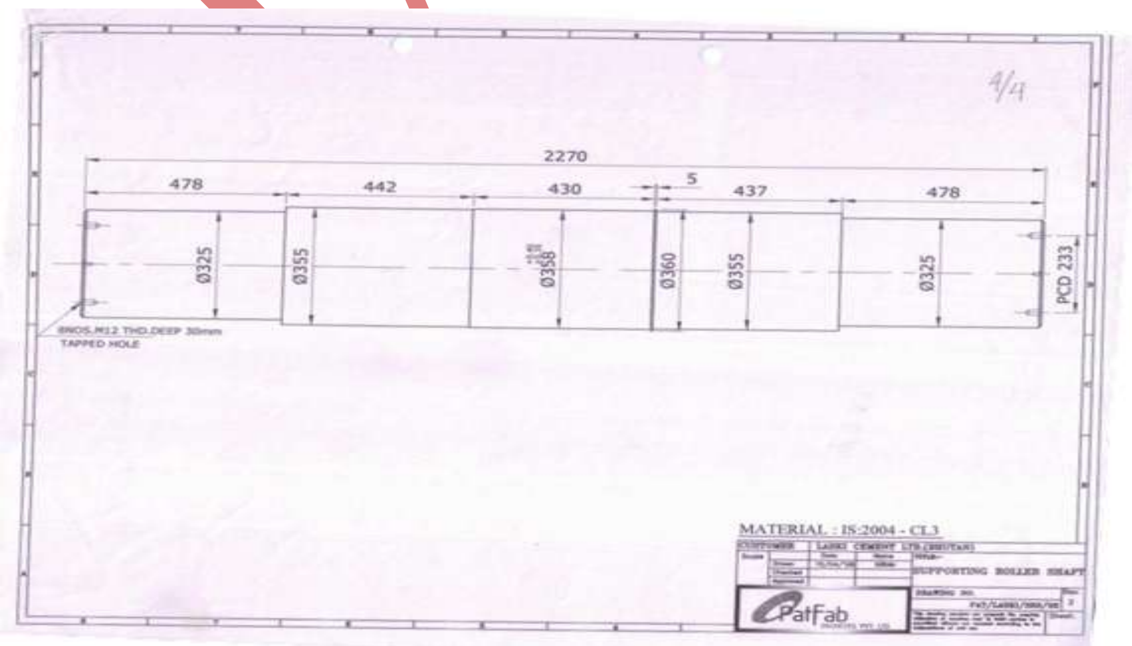


Deformation after suggested modification (case 1)



Stresses after suggested modification (case 1)

Fig.5.7 Deformation After Suggested Modification **Fig5.8 Stresses After Suggested Modification**



VI. TABLES AND GRAPHS

Finite element modeling of the contact between two cylinders was examined in detail. The finite element method with special techniques, such as the incremental technique of applying the external load in the input file, the deformation of the stiffness matrix, and the introduction of the contact element were used. It was found that initial loading using displacements as inputs was helpful in reducing numerical instabilities

To determine the accuracy of the present method for the bending stresses, both three dimensional and two dimensional models were built in this chapter. So those FEA models are good enough for stress analysis.

VII. DISCUSSION

The contribution of the thesis work presented here can be summarized as follows: It was shown that an FEA model could be used to simulate contact between two bodies accurately by verification of contact stresses between two cylinders in contact and comparison with the Hertzian equations.

Correct Mechanical Balance can be concluded as an optimum state of the forces and stresses distribution acting on kiln carrying system and the shell. The essential factors which determine this distribution are kiln geometry (ovality) and mutual relations between rotation axes of kiln and support rollers (dynamic relations)

The analyses of the circumferential stresses and the contact stresses are implemented in the FE code ANSYS. Subsequently, the required tangential friction stress is obtained in terms of the rolling and sliding contact area condition. This study can be used to solve a fundamental contact problem similar to the roller. The fatigue life curve can provide basis to adjust the axis line deflection more effectively and prevent the accelerated fatigue damage of the roller. In comparison with the previous results, the present results are longer and more rational as the multiaxial stress condition including the circumferential stress and the tangential friction stress is considered

VIII. ACKNOWLEDGEMENTS.

I would like to express the special thanks to my respected Dr. Mukul Shukla (MNNIT Allahabad), Dr.Sunil Somani (Director Medicaps) and Management and staff at Allen Smith Engineers Ltd., Mumbai, who have appreciated me directly and indirectly.

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Figure

Figure 1. Cement Rotary Kiln Ring and Roller View

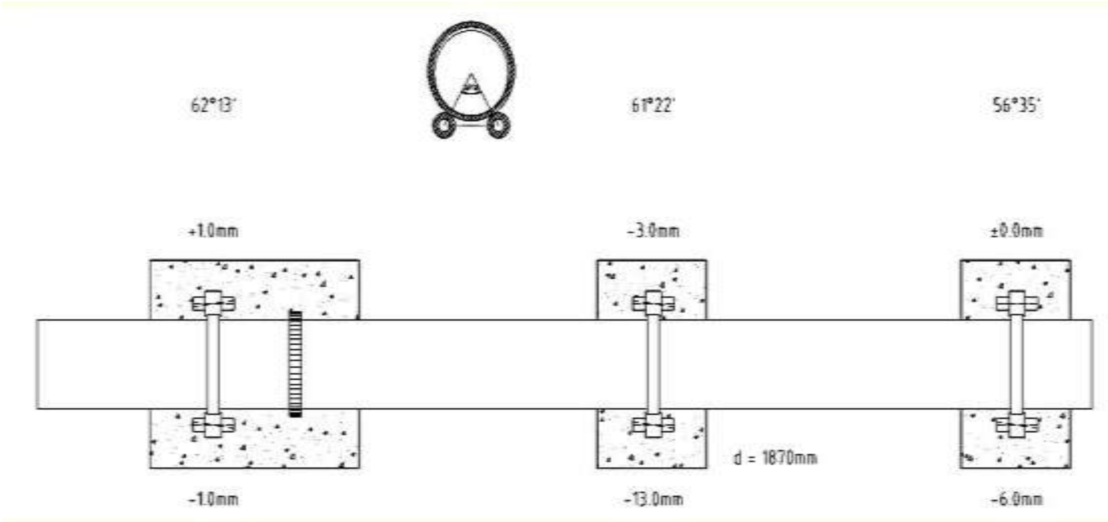

Fig. 3 Kiln Shell Profile After Multi Axial Deformations [Source: Geoservsex]

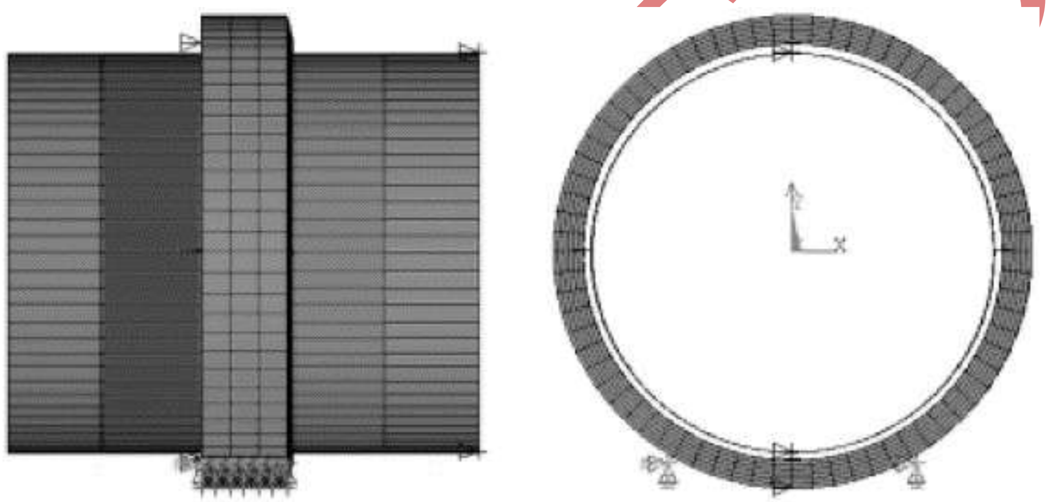

Fig 5 Finite Element Model Of Roller And Tires With Boundary Conditions [16]

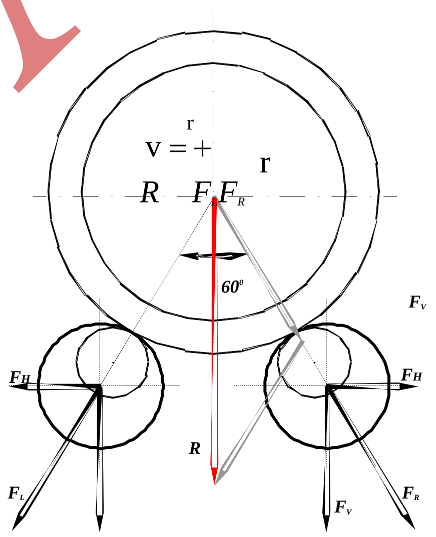

Fig.6 Scheme of Forces Distribution Sumptions and Additional Data Taken or Not Taken Into Account

Table 0.1

Gear Pitch Circle Diameter (m)	Girth Gear Axial Runout		Girth Gear Radial Runout	
	new installations (mm)	old installations (mm)	new installations (mm)	old installations (mm)
6.0	0.50	0.96	0.74	1.40
6.5	0.56	1.14	0.81	1.52
7.0	0.61	1.32	0.90	1.75