THE INFLUENCE OF UNBALANCED MAGNETIC PULL ON THE RELIABILITY OF INDUCTION MOTORS.

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TO

my wife, Dulcie

and children, Berdene and Samara.

SYNOPSIS.

This thesis is a study of the influence of unbalanced magnetic pull (u.m.p.) and associated phenomena on the reliability of induction motors.

The work arose from investigations into the premature failure or malfunction of several large and important 6600 volt induction motors whilst in service.

Insofar as the writer can determine much of the work reported in this thesis constitutes a new contribution to the experimental and theoretical knowledge of induction motors. This applies particularly to the experimental observations on large 4-pole induction motors, to the theoretical analysis and discussions and to the development of detailed enquiry documents and associated documents which include all the principal factors involved in the selection of 6600 volt induction motors.

The success of the writer's approach has been shown by the outstanding performance and perfect reliability in service of all the motors selected on this basis. This supports the writer's thesis that the effects of the radial electromagnetic forces, including the unbalanced magnetic pull due to airgap eccentricity must be carefully taken into account in the design and construction of induction motors in order to achieve high reliability.

The following preface provides an expanded abstract of this work.

PREFACE.

This thesis is a study of the influence of unbalanced magnetic pull (u.m.p.) and associated phenomena on the reliability of induction motors.

The work arose from the writer's investigations into the premature failure or malfunction of several large and important 6600 volt industrial induction motors whilst in actual service, the troubles concerned being cases of motor bearing failure, cases of rotor rubbing on stator and cases of excessive vibration.

This thesis includes the following:-

1. An introduction giving a brief history of the subject and describing how the writer became involved in this study.

2. A literature survey.

3. A note on failures of large induction motors experienced by the writer.

4. Detailed reports of experimental observations on large 4-pole induction motors whilst in operation on site and on test bed.

   These measurements have revealed the following principal phenomena:-

   4.1 Pulsations of vibration amplitude related to slip frequency.

   4.2 Changes in the value of vibration amplitude and changes in the pulsation of vibration with position on the motor bearings and motor frame.

   4.3 Pulsations of stator current related to slip frequency.

   4.4 Unusual variations of rotor current.

   Test bed observations, including the measurements of critical speed curves, were made for different supply voltages and frequencies and for different values of airgap eccentricity.

   The writer's work with others on the measurement of unbalanced magnetic pull in a large industrial motor is briefly reported upon.

   Insofar as the writer can ascertain, several of the phenomena observed experimentally have not been reported on before.

5. The following theoretical studies and discussions:-

5.1 A detailed theoretical analysis of the electromagnetic and mechanical radial loading forces present in induction motors. The subject has been dealt with broadly with concentration on the electromagnetic forces due to airgap eccentricity, particularly the unbalanced magnetic pull and vibratory forces. Insofar as the writer can determine this study provides the most explicit description of the phenomena to date and the mathematical theory is the most advanced.

5.2 Theoretical studies and discussions on the mechanical and structural load carrying capacity of induction motor components including the loading of ball and/or roller bearings, the loading of journal bearings and the loading of stator assemblies.

5.3 Theoretical studies and discussions on the effects of a squirrel cage rotor on the unbalanced magnetic pull, the effects of parallel paths in the stator winding on u.m.p., pulsations of vibration at slip frequency, critical speed curves.

6. A study of the design practices of several different motor manufacturers with respect to their methods of calculating unbalanced magnetic pull and the application of the value of unbalanced magnetic pull so calculated in the design of induction motors.

7. A discussion on the writer's development of the following documents, which documents have been included in this thesis:-

7.1 Enquiry Documents for 6600 volt Induction Motors, comprising:-

7.1.1 Introduction to Documents.

7.1.2 Form of Enquiry.

7.1.3 Schedule of Conditions.

7.1.4 Specification.

7.1.5 Form of Tender.

7.2 Schedule of Information Required from Supplier of Machinery to be Driven.

These documents include all the principal factors involved in the selection of 6600 volt induction motors and therefore go well beyond the scope of this thesis.

A discussion is included on the analysis of tenders submitted in response to enquiries for 6600 volt induction motors. The analysis of tenders has gone hand in hand with the development of the abovementioned documents as well as with the study of unbalanced magnetic pull and associated phenomena.

8. Conclusions, as follows.

Insofar as the writer can determine much of the work reported in this thesis constitutes a new contribution to the experimental and theoretical knowledge of induction motors.

This applies particularly to the experimental studies of vibration, stator current and rotor current, to the theoretical analysis of the electromagnetic forces, including unbalanced magnetic pull, and of the mechanical loading forces in induction motors, to the discussions on the loading of induction motor components and to the development of detailed enquiry documents and associated documents which include all the principal factors involved in the selection of 6600 volt induction motors.

The success of the writer's approach and of these documents in particular has been shown by the fact that all the motors which have been selected in terms of these documents and which have been put into service have demonstrated outstanding performance and perfect reliability. This dramatic reversal of the unfortunate trend which initiated the investigation in the most concrete support of the writer's thesis that the effects of the radial electromagnetic forces, including the unbalanced magnetic pull due to airgap eccentricity, must be carefully taken into account in the design and con-struction of induction motors in order to achieve high reliability.

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PROFESSOR N.C. ENSLIN of the Department of Electrical University of Cape Town, whose stimulating interest in my practical industrial problems encouraged me to investigate these problems more deeply. This initiated the idea of undertaking this thesis project. My grateful thanks to Professor Enslin for his encouragement and tactful supervision during the course of this project and for his painstaking concurrent work in the laboratory.

My special thanks to my wife for her loving encouragement and for her unstinting assistance. She and my children are to be admired for their patience and understanding and for the sacrifices they gladly made in the interest of this project.

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VOLUME 2.

These documents incorporate their own detailed tables of contents with cross-references between component documents.

These are arranged in chronological order. (136 PAGES)

LIST OF PRINCIPAL SYMBOLS.

a = area of airgap.

a = component of amplitude in the general equation for sinusoidal modulation.

= $y_1 y_3$

b = component of amplitude in the general equation for sinusoidal modulation.

= $y_2 y_3$

A = constant representing any constant term of $B^2_{\theta,t}$

$b(x,t)$ = flux density at angle x and at time t in FREISE and JORDAN's analysis.

= $B_{\theta,t}$

B = airgap flux density between two surfaces.

B_1 = amplitude factor in FREISE and JORDAN's expressions for flux density.

= $-\frac{B_p}{u}$

B_p = amplitude of airgap flux density of fundamental field with p pole-pairs.

= $M_1 \Lambda_0$

B_p = fundamental field with p pole-pairs in VON KAEHNE's survey.

B_p has this meaning in some of RAI's expressions. When B_p has this meaning it is the same as $\bar{B}_p$ in the writer's analysis.

B_{p+1} = airgap flux density of harmonic field (eccentricity field) having (p + 1) pole-pairs in RAI's expressions.

= $B_{\epsilon 1}$ = first harmonic field with (p + 1) pole-pairs in VON KAEHNE's survey.

= $\bar{B}_{p+1}$ in the writer's analysis.

B_{p-1} = airgap flux density of harmonic field (eccentricity field) having (p - 1) pole pairs in RAI's expressions.

= $B_{\epsilon 2}$ = second harmonic field with (p - 1) pole-pairs in VON KAEHNE's survey.

= $\bar{B}_{p-1}$ in the writer's analysis.

$\bar{B}_p$ = fundamental field

= $B_p \cos(p\theta - \omega t - \phi)$

$\bar{B}_{p+1}$ = eccentricity field (harmonic field) having (p + 1) pole-pairs.

$= \frac{\epsilon}{2} B_p \cos\left\{(p+1)\theta - (\omega + \omega_\epsilon)t - (\phi + \phi_\epsilon)\right\}$

$\bar{B}_{p-1}$ = eccentricity field (harmonic field) having (p - 1) pole-pairs.

$= \frac{\epsilon}{2} B_p \cos\left\{(p-1)\theta - (\omega - \omega_\epsilon)t - (\phi - \phi_\epsilon)\right\}$

$B_{\theta,t}$ = flux density at angle θ and at time t

$B'_{\epsilon 1}$ = first harmonic field with (p + 1) pole-pairs in VON KAEHNE's survey

$= B_{p+1}$

$= \bar{B}_{p+1}$

$B_{\epsilon 2}$ = second harmonic field with (p - 1) pole-pairs in VON KAEHNE's survey.

$= B_{p-1}$

$= \bar{B}_{p-1}$

C = basic dynamic load rating of bearing, that is, its carrying capacity in force units for a nominal life of 1 million revolutions (1 Mr). Used in life formula for ball and roller bearings.

C_0 = basic static load rating of ball and/or roller bearing.

d = radial deflection of stator frame in ALGER's expression, inch

D = diameter of stator bore

$= 2R$, metres

$= 2r$ in BRADFORD's expressions.

D_S = mean diameter of stator core, in inches. Used in ALGER's expression for radial deflection of stator frame.

D_V = Vertical deflection, that is, change in the vertical diameter due to loading of a ring by two opposing loads at diametrically opposite points on a vertical line. Used in ROARK and YOUNG's expressions.

D_H = horizontal deflection, that is, change in the horizontal diameter due to loading of a ring by two opposing loads at diametrically opposite points in a vertical line. Used in ROARK and YOUNG's expressions.

e = displacement between stator and rotor, specifically, the eccentric displacement of the centre of the rotor from the centre of the stator, as defined in Fig. 1

e = total eccentricity when considered as the resultant of the vectorial addition of e_0 and $e_1 \cos \omega_f t$ as defined in Fig. 3.

e_0 = stationary (non-oscillating) eccentricity and rotating eccentricity when considered as a vectorial component of the total eccentricity e, as defined in Fig. 3.

e_1 = amplitude of horizontally oscillating but non-rotating eccentricity when considered as a vectorial component of the total eccentricity e, as defined in Fig. 3.

E = YOUNG's modulus of elasticity.

f_e = one-sided magnetic pull in HELLER and HAMATA's expressions.

= F_u

F_{max} = absolute maximum load on ball and/or roller bearing.

F = attractive force per unit area

$F_{\theta,t}$ = attractive force per unit area at angle θ and time t

F_u = unbalanced magnetic pull in RAI's expressions

= f_e

F_{u1} = unbalanced magnetic pull for $p \neq 1$ in RAI's expressions.

F_{u2} = unbalanced magnetic pull for $p = 1$ in RAI's expressions.

$F(\epsilon)_{p\pm1}$ = function of eccentricity for the eccentricity fields having pole-pairs (p $\pm$ 1) in RAI's expressions, where

$$B_{p\pm1} = B_p \, F(\epsilon)_{p\pm1}$$

h = radial depth of stator core behind the slots, in inches. Used in ALGER's expression for radial deflection of stator frame.

I = moment of inertia of ring cross-section in ROARK and YOUNG's expressions.

I = moment of inertia of cross-section of stator frame in calculation of frame deflection.

= constant

= $\dfrac{\omega_t}{\omega_R}$

K = flexural rigidity of stator frame in calculation of frame deflection.

k_V = constant for determination of vertical deflection due to loading of a ring by two opposing loads at diametrically opposite points in a vertical line. Used in ROARK and YOUNG's expressions.

k_H = constant for determination of horizontal deflection due to loading of a ring by two opposing loads at diametrically opposite points in a vertical line. Used in ROARK and YOUNG's expressions.

l_e = effective length of core in FREISE and JORDAN's analysis.
= l

l = effective length of rotor.

L = bearing life in millions of revolutions when subjected to the bearing load, P force units. Used in life formula for ball and roller bearings.

L = length of stator frame in calculation of frame deflection.

m = factor for multiplying θ in the analysis of terms comprising $B^2_{\theta,t}$

M_1 = amplitude of m.m.f. wave form in the simple expression for m.m.f. assumed in the writer's analysis.

$-M_1$ = amplitude of m.m.f. wave form in FREISE and JORDAN's expression for $mmf_{\theta,t}$.

m_1 = amplitude of fundamental vibrational force of rotational frequency due to mechanical out of balance.

m_2 = amplitude of harmonic component of vibrational force of twice rotational frequency due to mechanical out of balance.

M_1^I = amplitude of foreward rotating component of m.m.f. wave form in MEILER, SPERLING and TIKVICKI's expression for $mmf_{\theta,t}$

M_2^I = amplitude of backward rotating component of m.m.f. wave form in MEILER, SPERLING and TIKVICKI's expression for $mmf_{\theta,t}$

M_1^{II} = amplitude of the fundamental m.m.f. wave form in the normal expression for $mmf_{\theta,t}$, where $mmf_{\theta,t}$ comprizes the fundamental, 5th, 7th, 11th, 13th .... harmonics.

M_5^{II} = amplitude of the 5th harmonic of m.m.f. wave form in the normal expression for $mmf_{\theta,t}$, where $mmf_{\theta,t}$ comprizes the fundamental 5th, 7th, 11th, 13th .... harmonics.

M_7^{II} = amplitude of the 7th harmonic of m.m.f. wave form in the normal expression for $mmf_{\theta,t}$, where $mmf_{\theta,t}$ comprizes the fundamental, 5th, 7th, 11th, 13th .... harmonics.

M_{11}^{II} = amplitude of the 11th harmonic of m.m.f. wave form in the normal expression for $mmf_{\theta,t}$, where $mmf_{\theta,t}$ comprizes the fundamental, 5th, 7th, 11th, 13th .... harmonics.

M_{13}^{II} = amplitude of the 13th harmonic of m.m.f. wave form in the normal expression for $mmf_{\theta,t}$ where $mmf_{\theta,t}$ comprizes the fundamental, 5th, 7th, 11th, 13th .... harmonics.

M_{1}^{III} = amplitude of the fundamental foreward revolving m.m.f. wave form in the general expression for $mmf_{\theta,t}$, where $mmf_{\theta,t}$ contains all harmonics.

M_{2}^{III} = amplitude of the second harmonic foreward revolving m.m.f. wave form in the general expression for $mmf_{\theta,t}$, where $mmf_{\theta,t}$ contains all harmonics.

M_{3}^{III} = amplitude of the third harmonic foreward revolving m.m.f. wave form in the general expression for $mmf_{\theta,t}$ where $mmf_{\theta,t}$ contains all harmonics.

M_{1}^{IV} = amplitude of the fundamental backward revolving m.m.f. wave form in the general expression for $mmf_{\theta,t}$, where $mmf_{\theta,t}$ contains all harmonics.

M_{2}^{IV} = amplitude of the second harmonic backward revolving m.m.f. wave form in the general expression for $mmf_{\theta,t}$, where $mmf_{\theta,t}$ contains all harmonics.

M_{3}^{IV} = amplitude of the third harmonic backward revolving m.m.f. wave form in the general expression for $mmf_{\theta,t}$, where $mmf_{\theta,t}$ contains all harmonics.

$mmf_{\theta,t}$ = magnetomotive force at angle θ and at time t

M_s = weight of stator core and windings + u.m.p. used in calculation of deflection of stator frame.

n = factor for multiplying θ in the analysis of terms comprising $B^{z}_{\theta,t}$

n = frequency ratio of P_n to P_1
Used in SHAWKI and FREEMAN's expression for the vertical cyclic load on a complete journal bearing.

0 = factor for the reduction of the fundamental field during the starting period due to primary reactances. This factor is used in FREISE and JORDAN's analysis.

$\approx$ 0,5 during starting.

$\approx$ 1,0 at synchronous speed.

p = number of pairs of poles.

p = pole-pair number of the machine.

p = exponent used in life formula for ball and roller bearings.

= 3 for ball bearings

= $\frac{10}{3}$ for roller bearings

p_1 = number of pole-pairs in the first of two harmonic fields which produce a force wave and deformations of rotor and stator.
Used in VON KAEHNE's survey.

p_2 = number of pole-pairs in the second of two harmonic fields which produce a force wave and deformations of rotor and stator.
Used in VON KAEHNE's survey.

P = bearing load (equivalent bearing load), in force units.
Used in life formula for ball and roller bearings.

P = load applied per unit projected bearing area.
Used in SHAWKI and FREEMAN's expression for vertical cyclic load on a complete journal bearing.

P = number of pole-pairs in ALGER's expression for radial deflection of stator frame.

P_0 = static equivalent load on ball and/or roller bearing.

P_0 = steady component of P.
Used in SHAWKI and FREEMAN's expression for the vertical cyclic load on a complete journal bearing.

P_1 = fundamental component of P.
Used in SHAWKI and FREEMAN's expression for the vertical cyclic load on a complete journal bearing.

P_n = nth harmonic component of P.
Used in SHAWKI and FREEMAN's expression for the vertical cyclic load on a complete journal bearing.

q = amplitude of component sinusoidal wave in the general expression for interference or beating.

R = radius of stator bore

R = radius of a ring subjected to two opposing loads at diametrically opposite points in a vertical line.
Used in ROARK and YOUNG's expressions.

r = radius of rotor bore.

r = radius of stator bore in RAI's expressions.

r = order of force waves producing deformations of rotor and stator.

= $\Delta p = p_1 \pm p_2 = p(\nu \pm \mu)$

Used in VON KAEHNES survey.

s_0 = safety factor of ball and/or roller bearing for static loading.

= $\frac{C_0}{P_0}$

$\geqq$ 0,5 for smooth shock free operation.

$\geqq$ 1 for ordinary service.

$\geqq$ 2 for sudden shocks and for high requirements on the smooth running of the bearing.

t = time

UMP = contribution of the "unbalanced magnetic pull" to the loading of a particular bearing.

u, v = factors depending upon eccentricity in FREISE and JORDAN's formula for $\Lambda_{\theta, t}$

FREISE and JORDAN gave the values of u and v in a graph showing their dependence on ϵ. For $\epsilon = 0$, $u = v = 1$. Both u and v increased more than linearly as ϵ increased. RAI reproduced this graph in his thesis.

w_1 = factor for the reduction of the harmonic field (p + 1) by rotor currents. Used in FREISE and JORDAN's analysis.

$\leqq$ 1,0 for squirrel cage motors.

$\approx$ 1,0 for slipring motors.

w_2 = factor for the reduction of the harmonic field (p - 1) by rotor currents. Used in FREISE and JORDAN's analysis.

$\leqq$ 1,0 for squirrel cage rotors.

$\approx$ 1,0 for slipring rotors.

W = contribution of weight of complete rotor to the loading of a particular bearing. This weight includes any external loads on the shaft.

W = value of each of two opposing loads on a ring at diametrically opposite points in a vertical line. Used in ROARK and YOUNG's expressions.

x = angle between x-axis and any point on the periphery of the stator bore, radians. Used in FREISE and JORDAN's analysis.

= θ

y = function expressing sinusoidal modulation.

y_1 = amplitude of the first component in the general equation for sinusoidal modulation.

y_1 = first sinusoidal wave producing interference or beating.

y_2 = amplitude of the second component in the general equation for sinusoidal modulation.

y_2 = second sinusoidal wave producing interference or beating.

y_3 = amplitude of the third component in the general equation for sinusoidal modulation.

α = angle between the centre line of eccentricity and any point on the periphery of the stator bore radians.

= $\theta - \omega_\epsilon t - \phi$, as defined in Fig. 1.

β = any angle which is a function of two or more of the angles ωt, $\omega_\epsilon t$, ϕ , ϕ_ϵ (but not of the angle θ) in the terms of $B^2_{\theta,t}$

Used in the analysis to find u.m.p.

γ = $\omega_\epsilon t + \phi_\epsilon$

γ is independent of θ
Used in the analysis to find u.m.p.

δ = mean airgap length

= $\dfrac{\delta_{max} + \delta_{min}}{2}$

δ = deflection of stator frame

= $\dfrac{M_s}{K}$

δ_{min} = minimum airgap length

δ_{max} = maximum airgap length

$\delta_{\theta,t}$ = airgap length at an angle θ and at time t

Δp = $p_1 \pm p_2 = p(\nu \pm \mu) = r$

Used in VON KAEHNE's survey.

$\Delta\omega$ = $\omega_4 - \omega_3$ in the general expression for interference or beating.

ϵ = per unit eccentricity of the airgap.

= $\dfrac{\text{displacement of stator and rotor centres}}{\text{average mechanical airgap length}}$

= $\dfrac{e}{\delta}$

= ε

ϵ = total per unit eccentricity when considered as the resultant of the vectorial addition of ϵ_0 and $\epsilon_1 \cos \omega_f t$, as defined in Fig. 3.

ϵ_0 = $\frac{e_0}{\delta}$ = per unit stationary (non-oscillating) eccentricity and rotating eccentricity when considered as a vectorial component of the total per unit eccentricity ϵ , as defined in Fig. 3.

ϵ_1 = $\frac{e_1}{\delta}$ = per unit amplitude of horizontally oscillating but non-rotating eccentricity when considered as a vectorial component of the total per unit eccentricity ϵ, as defined in Fig. 3.

θ = angle between the x-axis and any point on the periphery of the stator bore, radians, as defined in Fig. 1.

$\Lambda_{\theta,t}$ = airgap permeance at angle θ and at time t

= $\frac{1}{\delta_{\theta,t}}$ in FREISE and JORDAN's analysis, also repeated by RAI.

= $\frac{\mu_0 a}{\delta_{\theta,t}}$ in the writer's analysis.

Λ_0 = permeance with nominal airgap

= $\frac{1}{\delta}$ in FREISE and JORDAN's analysis, also repeated by RAI.

= $\frac{\mu_0 a}{\delta}$ in the writer's analysis

Λ_{01} = factor depending upon eccentricity in FROHNE's formula for $\Lambda_{\theta,t}$

= $\frac{1}{\sqrt{1-\epsilon^2}} \doteqdot \frac{1}{1-\frac{\epsilon^2}{2}}$ in the range $\epsilon < 0,7$

Λ_1 = factor depending upon eccentricity in FROHNE's formula for $\Lambda_{\theta,t}$

= $\frac{2(1-\sqrt{1-\epsilon^2})}{\epsilon\sqrt{1-\epsilon^2}} \doteqdot \frac{\epsilon}{1-\frac{\epsilon^2}{2}}$ in the range $\epsilon < 0,7$

Λ_2 = factor depending upon eccentricity in FROHNE's formula for $\Lambda_{\theta,t}$

= $\frac{2(1-\sqrt{1-\epsilon^2})^2}{\epsilon^2\sqrt{1-\epsilon^2}}$

Λ_n = factor depending upon eccentricity in FROHNE's formula for $\Lambda_{\theta,t}$

= $\frac{2(1-\sqrt{1-\epsilon^2})^n}{}$

μ_0 = permeability of free space.

= $4\pi \times 10^{-7}$ H/m

μ_r	=	relative permeability.
μ	=	$\frac{p_2}{p}$ in VON KAEHNE's survey.
ν	=	order of the harmonic radial magnetic field.
ν	=	$\frac{p_1}{p}$ in VON KAEHNE's survey. This meaning differs from the first definition given above.
ϕ_{m1}	=	phase angle of fundamental vibrational force of rotational frequency due to mechanical out of balance.
ϕ_{m2}	=	phase angle of harmonic component of vibrational force of twice rotational frequency due to mechanical out of balance.
ϕ_ε	=	ϕ_ϵ
ϕ_ϵ	=	phase angle of permeance wave, radians
	=	angle between the x -axis and the centre of the line of eccentricity when $t=0$, as defined in Fig. 1
	=	ϕ_ε
ϕ	=	phase angle of m.m.f. wave, electrical radians.
	=	angle between the x -axis and the centre of the magnetic pole when $t=0$, as defined in Fig. 1.
ϕ'	=	phase angle of m.m.f. wave, electrical radians. Used in FREISE and JORDAN's analysis.
	=	ϕ
ψ	=	phase relation between P_n and P_1.
		Used in SHAWKI and FREEMAN's expression for the vertical cyclic load on a complete journal bearing.
ω	=	angular velocity, radians/s
	=	angular velocity corresponding to supply frequency.
ω_f	=	angular velocity applicable to variation in horizontal displacement between the rotor and the stator axis in LANDY's formula for $\delta_{\theta,t}$
ω_R	=	angular velocity of rotor, rad/s
ω_1	=	angular speed of application of P_1.
		Used in SHAWKI and FREEMAN's expression for the vertical cyclic load on a complete journal bearing.
ω_ϵ	=	angular velocity corresponding to the frequency of rotating eccentricity, rad/s
ω_ε	=	ω_ϵ

ω_2 = angular velocity corresponding to frequency of the second component in the general equation for sinusoidal modulation.

ω_3 = angular velocity corresponding to frequency of the third component in the general equation for sinusoidal modulation.

ω_3 = angular velocity corresponding to frequency of the first component sinusoidal wave y_1 , in the general expression for interference or beating.

ω_4 = angular velocity corresponding to frequency of the second component sinusoidal wave y_2 in the general expression for interference or beating.

1. INTRODUCTION.

The unbalanced magnetic pull (u.m.p.) arising from airgap eccentricity is known to have an adverse effect on the reliability of induction motors.

Considerable attention has been paid to the study of this subject since at least the first decade of this century.

In 1963 the Electrical Research Association in the United Kingdom (ERA) published a survey by VON KAEHNE[49] of published work on unbalanced magnetic pull which disclosed many inconsistencies in the various theories and approaches. These theories were not well supported by experimental data. As a result a research project was begun at the ERA to investigate critically the adverse effects and to establish a consistent theory. This work was reported in 1968 by BRADFORD[5][6] and in 1973 by RAI.[33] RAI claimed that, as a result of this work a simple but accurate theory to calculate u.m.p. and vibratory forces had been developed and that these forces could be predicted at the design stage.

However, in 1973, BINNS and DYE[4] reported the results of their investigations. RAI[4] stated that the conclusions of BINNS and DYE disagreed radically with published data and extensive measurements made at the ERA.

In the meanwhile the writer has found serious errors in RAI's theory and that RAI's basic formulae for u.m.p. conflict with the formulae published by FREISE and JORDAN (1962),[23] by BRADFORD (1968),[5][6] by MEILER, SPERLING and TIKVICKI (1973)[31] and by HELLER and HAMATA (1977).[26] The writer's detailed investigations in this field are given in this thesis.

The writer has, from time to time during the course of his duties over a period of about 25 years, been involved in the investigation of a wide variety of motor failures. From about 1970 these motor failures intensified in a particular direction and the writer became deeply involved in investigations of this particular type of motor failure. Specifically, these motor failures involved several different cases of premature failure or malfunction of various large and important 6600 volt industrial motors whilst in actual service, the troubles concerned being cases of motor bearing failures, cases of rotor rubbing on stator and cases of excessive vibration sometimes leading to cracking of the stator frames. All the motors involved had been purchased in the late 1960's and early 1970's and had given little service before giving trouble.

These investigations in the course of time revealed a complexity of problems involving electromagnetic, mechanical and structural considerations. The influence of unbalanced magnetic pull on the reliability of induction motors was the question which predominated in these investigations. This led to an increasing interest in the study of the u.m.p. arising from airgap eccentricity, and this, in turn, led to a deeper study of the complex electromagnetic and mechanical forces present in induction motors and their effect on the reliability of these motors.

The abovementioned investigations arose from the writer's duties as an Engineer employed by Union Corporation Limited, a leading South African mining finance house based in Johannesburg.

Union Corporation Limited controls a number of important South African industries, principally large gold mines and large platinum mines with their associated metallurgical extraction plants, as well as the largest South African pulp and paper manufacturing industry. In addition Union Corporation has interests in several other important industries in South Africa and other countries. Union Corporation provides various types of managerial and consulting engineering services to the abovementioned industries.

These industries use a wide range of sizes and types of electric motors. The writer's viewpoint is therefore that of a user of motors rather than that of a designer or manufacturer of motors.

Previously published work on the subject of the adverse effects of unbalanced magnetic pull has often had academic origins. However, much work has been done from the viewpoint of the motor designer and manufacturer, although the work itself may have been done in a university or research organization. For example, the abovementioned research work at the ERA was sponsored by the leading U.K. and Continental manufacturers of electrical machines. The progress of the work was reviewed continuously by the sponsors. In this way the work was considered to be closely aligned to the needs of the motor manufacturing industry.

On the other hand, there appears to be little, if any, significant published scientific contribution by users, although there is ample evidence, some published[22], of users'[28][48] concern with the question of reliability. The writer's attempts, as a user, to look more closely at the problems involved and apply scientific method to their investigation may be almost if not entirely unique.

As a result of his investigations into the abovementioned problems, the writer was invited to represent the Chamber of Mines of South Afica on the Co-ordinating Committee on Large Rotating Machines of the Council for Scientific and Industrial Research (CSIR), National Electrical Engineering Research Institute (NEERI) from the 22nd August, 1975. This committee was later re-organised as the Rotating Machines Working Group of the High Voltage Co-ordinating Committee, CSIR,NEERI. The meetings of this working group have indicated a continuing interest in the subject.

Professor N.C. ENSLIN represented the University of Cape Town on the Co-ordinating Committee and at present represents the University of Cape Town on the Working Group. This led to a fairly continuous exchange of information and views between Professor Enslin and the writer from August, 1975 up to the present. This has stimulated the writer to investigate the subject more deeply than he would otherwise have done, particularly the making of detailed measurements on large motors whilst in service and on the test bed and the investigation of the theoretical aspects of the subject.

This work has led to the presentation of this thesis.

2. LITERATURE SURVEY.

A survey of published work on unbalanced magnetic pull in rotating machines was made by VON KAEHNE in 1963. [49] He included a list of 64 references.

RAI's thesis in 1973 [33] including a review of published work in connection with the effect of airgap eccentricity in an induction machine. A list of 71 references was provided which covered work on this effect and related phenomena.

HELLER and HAMATA's book in 1977 [26] included a list of 308 references in connection with harmonic field effects and related phenomena in induction machines.

ALGERS' book in 1970 [2] provided a list of references with each chapter. Because this is a general text on induction motors the references applicable to the chapters dealing with the effects produced by harmonic fields are relevant to the present study.

The writer has avoided reproducing the surveys, reviews and lists of references that have already been well provided in the abovementioned works. At the same time the field covered by this thesis is different to that previously considered by others although there are overlapping areas of interest. Therefore the list of references provided in this thesis includes types of literature not considered by the abovementioned investigators in addition to including specific items of literature listed by them.

The list of references provided in this thesis is essentially literature which the writer has referred to in his studies and which is relevant to the subject of this thesis, but this list is not intended as an exhaustive list of literature on the subject.

The relevant past work is referred to in this thesis at appropriate points in the study of the theory as well as in the discussions on the results of the experimental observations.

3. FAILURES OF LARGE INDUCTION MOTORS.

As indicated elsewhere in this thesis, the writer has, over a period of several years, been involved in investigations into the premature failure of large and important 6600 volt industrial induction motors.

These investigations stimulated the writer's interest in the field of research forming the subject of this thesis.

In several cases, after the problems were studied repairs and modifications were made but no formal reports were written. In other cases the writer himself, or engineers working under the guidance of the writer, wrote detailed reports on these problems. Copies of these reports have been submitted to the University. Space limitations make it impractical to include these reports in this thesis.

Failures of large induction motors have been mentioned in several places in this thesis. Reference should be made particularly to the reports of the writer's experimental observations included in this thesis. All the motors on which these experimental observations were made suffered premature failures whilst in service. For further information of the writer's experiences of premature failures of large induction motors, reference should be made to the three articles by the writer published in the Research Review, Department of Electrical Engineering, University of Cape Town. [8][9][10]

4. EXPERIMENTAL OBSERVATIONS.

4.1 INTRODUCTORY REMARKS:

The experimental observations reported in this thesis were made on the following motors:-

1. Five nominally identical 1800 kW, 4 pole, 6600 volt slip-ring induction motors, each intended to drive a centrifugal high-lift mine-dewatering pump. These motors were fitted with ball and roller bearings. The first of these motors was installed in the last quarter of 1975, well over a year before the tests reported in this thesis were conducted. This motor displayed extremely high vibration. Various measures were taken to correct conditions that were thought to cause or contribute to the vibration. Some of these measures were successful in reducing the vibration to an extent, but not down to normally acceptable limits under normal running conditions. As successive motors in this set were put into service they displayed similar high levels of vibration. Unfortunately circumstances demanded that these motors be operated despite their defective behaviour. Furthermore, it was found that the vibration of each motor progressively increased over a period of a few weeks or months, despite repeated corrective measures, until values of vibration of 500 to 1000 microns, peak to peak, were reached. If the motor had not already failed it was removed at this stage for complete overhaul and rebalancing.

   Various tests were done on these motors before the tests reported in this thesis. In these early tests the vibration was measured on the bearings only, in the horizontal, vertical and axial directions. No vibration readings were taken on the frame of the motor. A frequency analysis of the vibration indicated that the main component of vibration was at rotational speed with a smaller component at twice rotational speed. Other tests indicated that the vibration increased slightly with an increase in temperature under certain circumstances but not under other circumstances. Still other tests showed that the vibration increased with increase in voltage under certain circumstances but not under other circumstances. When a motor was overhauled and balanced and then tested in a workshop the vibration was brought down to very satisfactory limits. The same motor on being moved underground in the mine onto its bed, even uncoupled from the pump, displayed completely unacceptable vibration levels. Another motor was found to display similar instability in its vibration level when it was moved from one workshop to another workshop.

   Copies of some of the test reports relating to this period have been submitted to the University. Space limitations make it impractical to include these reports in this thesis.

   These motors eventually suffered bearing failures and the frames began to crack near the motor feet. In one case a rotor winding connection cracked open. One rotor winding developed a fault to earth probably due to cracked insulation and the ingress of carbon dust.

2. One 1800 H.P. (1340 kW), 4 pole, 6600 volt slipring induction motor intended to drive a centrifugal high lift mine dewatering pump.

This motor had oil lubricated sleeve bearings. These bearings suffered repeated premature failures each time involving damage to the shaft at the bearing journals.

In addition to the experimental observations reported in this thesis the writer participated with others in a programme aimed at measuring the unbalanced magnetic pull and associated phenomena present in this motor. Apart from the work done by others this project involved the writer in a great deal of time and effort. The project was initiated by the writer's queries regarding the value of u.m.p. present in this motor. It is intended that this work will be the subject of a report or paper to be written by the team who participated in the investigation. The writer has submitted to the University whatever records were available to him of this test work, his own discussions of the observations and his conclusions.

The original endshields of the motor were modified enabling the rotor to be set to various values of stationary eccentricity. Each endshield was cut into two pieces, that is, an inner ring, to which the bearing housing was fitted, and an outer ring, which was bolted to the stator frame. An arrangement of eight radial arms was used to connect the inner ring of each endshield to the outer ring of each endshield. One pair of radial arms was mounted at each of the following positions : 3 o'clock, 6 o'clock, 9 o'clock, 12 o'clock. For each pair of radial arms, one arm was mounted on the outside of the endshield rings and one arm was mounted on the inside of the endshield rings. The arms were so designed and dimensioned that by means of the use of resistance strain gauges on each arm it was possible to measure the force acting between the rotor and the stator at the bearings. The strain gauge measuring system was calibrated against a spring balance which measured the force applied by means of an overhead crane. The calibration forces were applied directly in the vertical direction and, via a deflecting pulley, in the horizontal direction.

Tests were first conducted with the motor stationary and the rotor open-circuited. Measurements of force (u.m.p.) against applied voltage (corresponding to magnetic flux density) were plotted for each of several values of eccentricity. Each such [29,p14]
curve was found to be of the form shown in Figure 10. VON KAEHNE (1963) pointed out that such a curve had been known since early in this century. He pointed out that the nominal value of magnetic flux density was usually above the value of the critical magnetic flux density, that is, the value of the magnetic flux [5][6]
density at the point of maximum u.m.p. BRADFORD (1968) measured a similar curve but found that the rated voltage (that is, the nominal value of magnetic flux density) was below the value of the critical voltage (that is, the critical magnetic flux density). In the case of the abovementioned tests conducted on the 1800 HP motor, the rated voltage was found to be well beyond the critical voltage. This means that at rated voltage the machine operated deeply into the saturated region. It was found that as the airgap eccentricity increased the peak of the u.m.p. vs. applied voltage curve increased in value.

Tappings were taken off the stator winding enabling the voltage to be measured across each phase band. By the use of a system of special voltage transformers and current [39] transformers it was possible to measure the KVAR resultant unbalance arising from the inequality of the magnetic field distribution around the circumference of the airgap. It was found that there was a correlation between the KVAR resultant unbalance measurements and the strain gauge measurements. For the maximum values of the u.m.p. vs. applied voltage curve there was very good correlation with the KVAR resultant unbalance measurements provided suitable scale adjustments were made to the latter measurements. The correlation was good and fairly consistent over the range of eccentricity measurements. The abovementioned scale adjustments having been made, correlation at 6,6 kV was not altogether satisfactory. The KVAR resultant unbalance measurements were comparatively low and not consistent over the range of eccentricity measurements. It is important to emphasize that this may not have been the fault of the KVAR resultant unbalance measurements but could well have been due to faulty strain gauge and/or eccentricity measurements.

At a later stage readings of u.m.p. vs. voltage were taken with the motor rotating at no-load. These readings were slightly higher than the measurements taken with the motor stationary. However, the accuracy of the measuring equipment was such that the values obtained with the motor rotating at no-load may be considered to have been essentially the same as those obtained for the motor stationary.

Later tests were done with the motor driving loads of different values at rated voltage and at a slightly lower voltage. The load tests showed that there was little, if any, significant increase in the steady u.m.p. with increase in load. There may even have been a decrease in the value of the steady u.m.p. with load. Unexpected measurement difficulties made this difficult to resolve but the results were certainly not in [33, pp 51-83, 90] agreement with RAI's experimental findings. RAI found that there was a significant increase in the u.m.p. as the load increased. RAI's experimental rig was the same as that used [5][6] by BRADFORD. It is possible that RAI found the u.m.p. to increase significantly with load because his machine was not operating as deeply into the saturated region as was the case in the abovementioned tests on the 1800 HP motor.

The tests on the 1800 HP motor showed that there was an increase in both the horizontal and vertical bearing vibration as the load increased.

The results of the tests on the 1800 HP motor confirm that the variation of u.m.p. with voltage and with eccentricity is of the same pattern as found by investigations on smaller motors except that the 1800 HP operates deeply into the saturated region and the variation of u.m.p. with load is not as would be expected from the results of tests on smaller laboratory motors. The vibration levels were found to be high increasing further with load. The results of these tests point to the need for further investigation of the phenomena in the saturated region.

The test reports which are included in this thesis are essentially as written after the respective tests were conducted. In the light of later experience and his own theoretical work, the writer could well have changed some of the wording but he has abstained from making such alterations.

A wealth of information is provided in these test results. Only some of the phenomena observed has been explained in the writer's theory. More theoretical work remains to be done to explain the observed phenomena.

4.2 TESTS ON 1800 KW, 4 POLE, 6600V SLIPRING INDUCTION MOTORS:

1800 KW, 4 POLE, 6600 V SLIPRING INDUCTION MOTOR DRIVING CENTRIFUGAL PUMP.

TESTS CONDUCTED AT LOWER PUMP STATION, UNDERGROUND, NO. 8 SHAFT, ST. HELENA GOLD MINES LIMITED ON MOTOR NO---82 INSTALLED IN NO.1 POSITION (NEAREST ENTRANCE TO PUMP STATION).

DATE OF TESTS: 22/2/77.

INSTRUMENT USED: IRD MECHANALYSIS PORTABLE VIBRATION METER MODEL NO. 306 M SERIAL NO. BO 8013641.

PLEASE NOTE ABBREVIATION: MU = μ = VIBRATION DISPLACEMENT READING, MICRONS PEAK TO PEAK.

CONDITION DURING TESTS:

The motor was coupled up to the pump. During the normal running tests in which vibration readings were taken on the bearing caps, motor frame, bedplate and foundation, the pump was pumping with its delivery valve fully open. Under these conditions motor stator current was 144 amps

NOTE: The rated full load current of the motor is 182 amps, so this is about 79.7 per cent of rated full load current.

VIBRATION READINGS ON BEARING CAPS:
(Stator Current 144 amps).

NON-DRIVE-END BEARING:

Horizontal: 54 to 65 MU. The needle of the vibration meter was oscillating regularly at 48 pulsations per minute, that is, counting from the peak of one oscillation of the needle to the peak of the next oscillation of the needle.

Vertical: 17 to 19 MU. The needle of the vibration meter was oscillating regularly at 24 pulsations per minute, as defined above.

Axial: 6 to 7 MU. The needle of the vibration meter was oscillating erratically. It was very difficult to define a distinct oscillation and to count the number of oscillations, but for the sake of giving a figure there could have been about 40 pulsations per minute.

DRIVE-END BARING:

Horizontal: 14 to 18 MU. The needle of the vibration meter was oscillating regularly at 48 pulsations per minute.

Vertical: 2 to 5 MU. The needle of the vibration meter was oscillating regularly at 24 pulsations per minute.

Axial: (These readings were taken above the shaft position on the bearing cap): 14 to 17 MU. The needle of the vibration meter was oscillating erratically. It was very difficult to define a distinct oscillation and to count the number of oscillations, but for the sake giving a figure there could have been about 55 pulsations per minute.

VIBRATION READINGS ON MOTOR FRAME, BEDPLATE AND FOUNDATION:
(Stator Current : 144 Amps).

POSITION	HORIZONTAL VIBRATION READING (MU)			
	DRIVE-END		NON-DRIVE-END	
	MINIMUM	MAXIMUM	MINIMUM	MAXIMUM
A	0.5	2	1	3
B	2	3	2	4
C	3	4	4.5	6
D	3	4	4	6
E	2.5	4	4.5	6.5
F	23	28	40	45
G	58	62	60	70
H	60	70	66	70
I	110	130	85	95
J	130	140	90	110

Please refer to diagram showing positions where readings were taken and graphical plot of readings.

The significance of these readings is that the vibration is held to a very low value in the foundation, bedplate and foot of the motor, but that the vibration increases very much as measurements are made progressively higher up on the motor frame itself. These facts contradict the argument of the motor manufacturer who has stated that the vibration is caused by an unsatisfactory bedplate and by unsatisfactory grouting of the bedplate.

The change from a low level of vibration to a high level of vibration occurs within the motor frame itself and not within the foundation, bedplate or foot of the motor. A sharp increase in the level of vibration within a structure must cause stress in the region of this sharp increase in level of vibration. This has been demonstrated by the cracks which have occurred in the motor from just above and below the gussets joining the feet of the motor to the barrel of the motor frame. (Positions F and E). This has already happened in at least two motors.

VIBRATION READINGS ON TOP EDGE OF MOTOR FRAME:
(Stator Current : 144 Amps)

POSITION	HORIZONTAL VIBRATION READING (MU)	
	MINIMUM	MAXIMUM
AA	18	20
BB	50	55
CC	60	70
DD	80	90
EE	85	95
FF	90	100
GG	80	90
HH	80	90

Please refer to diagram showing positions where readings were taken and graphical plot of readings.

Readings on the drive-end endshield at Positions AA and BB were taken on the hexagonal heads of the screws holding the ventilation screen. This endshield had been turned through 180° to direct the exhaust air upwards but this had not been done with the non-drive-end endshield. The air intake screen was at the bottom of the motor. Therefore there were no screw heads available on the top of the non-drive-end endshield, so it was not possible to measure the horizontal vibration on the top edge of the non-drive-end endshield.

These readings show that the vibration increased considerably as the readings were taken at positions progressively further from the drive-end bearing. The readings rise to a maximum at a position corresponding roughly to the centre of the magnetic core length. Progressing from this position to the non-drive-end the readings decrease slightly.

Earlier investigations on these motors were directed mainly to taking readings of vibration on the bearings. However the readings now taken on the frame of the motor show that the vibration is higher on the frame of the motor itself than on the bearings, that the vibration increases as readings are taken higher up on the motor frame (Please refer to readings of horizontal vibration in the vertical plane through the motor holding down bolts and each end of the motor), and that the vibration readings are higher nearer the magnetic core of the motor. This suggests that the vibration could arise from the magnetic attraction between the stator core and the rotor core and therefore that the vibration arises from unbalanced magnetic pull in the motor.

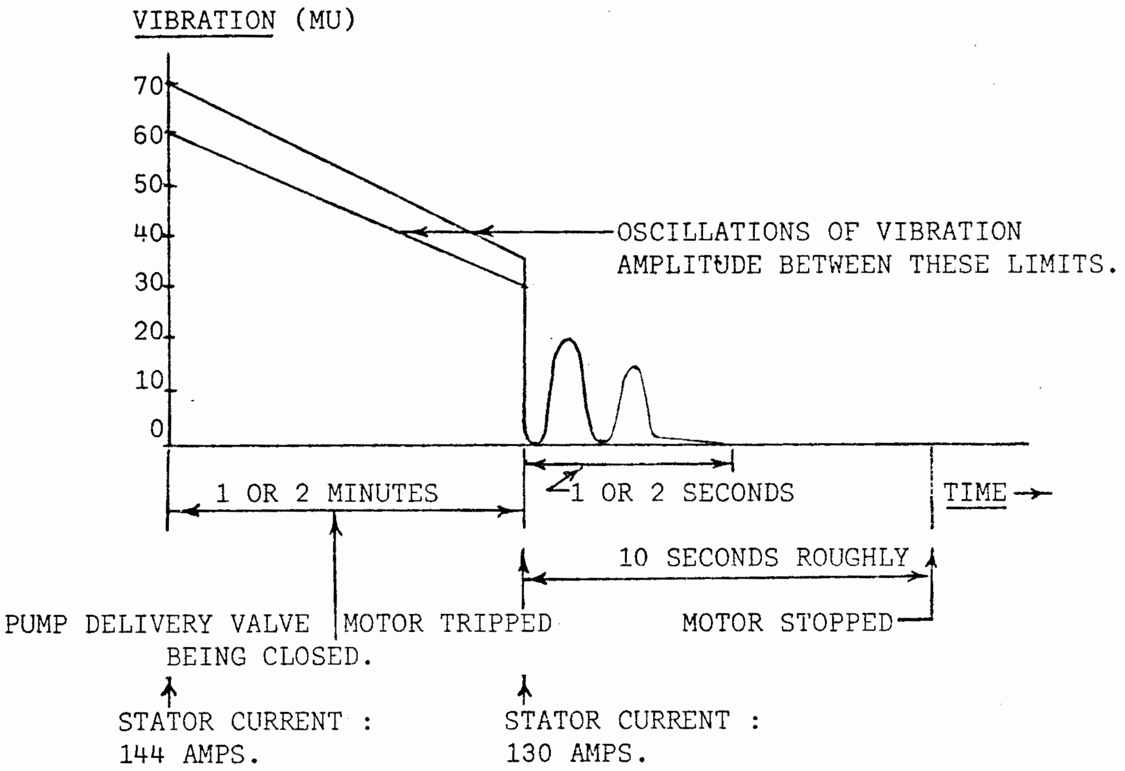
VIBRATION READINGS DURING STOPPING SEQUENCE:

These readings were taken on the non-drive-end bearing in the horizontal direction.

Running Condition with stator current 144 amps : 60 to 70 MU (oscillating).

The pump delivery valve was gradually closed. During this period the stator amps decreased progressively to 130 amps. The vibration decreased to a value of 30 to 35 Mu (oscillating). At this point the operators tripped the motor. The valve was not fully closed as it should have been. Immediately the motor tripped the vibration dropped to zero, then oscillated up and down twice and then decayed to zero.

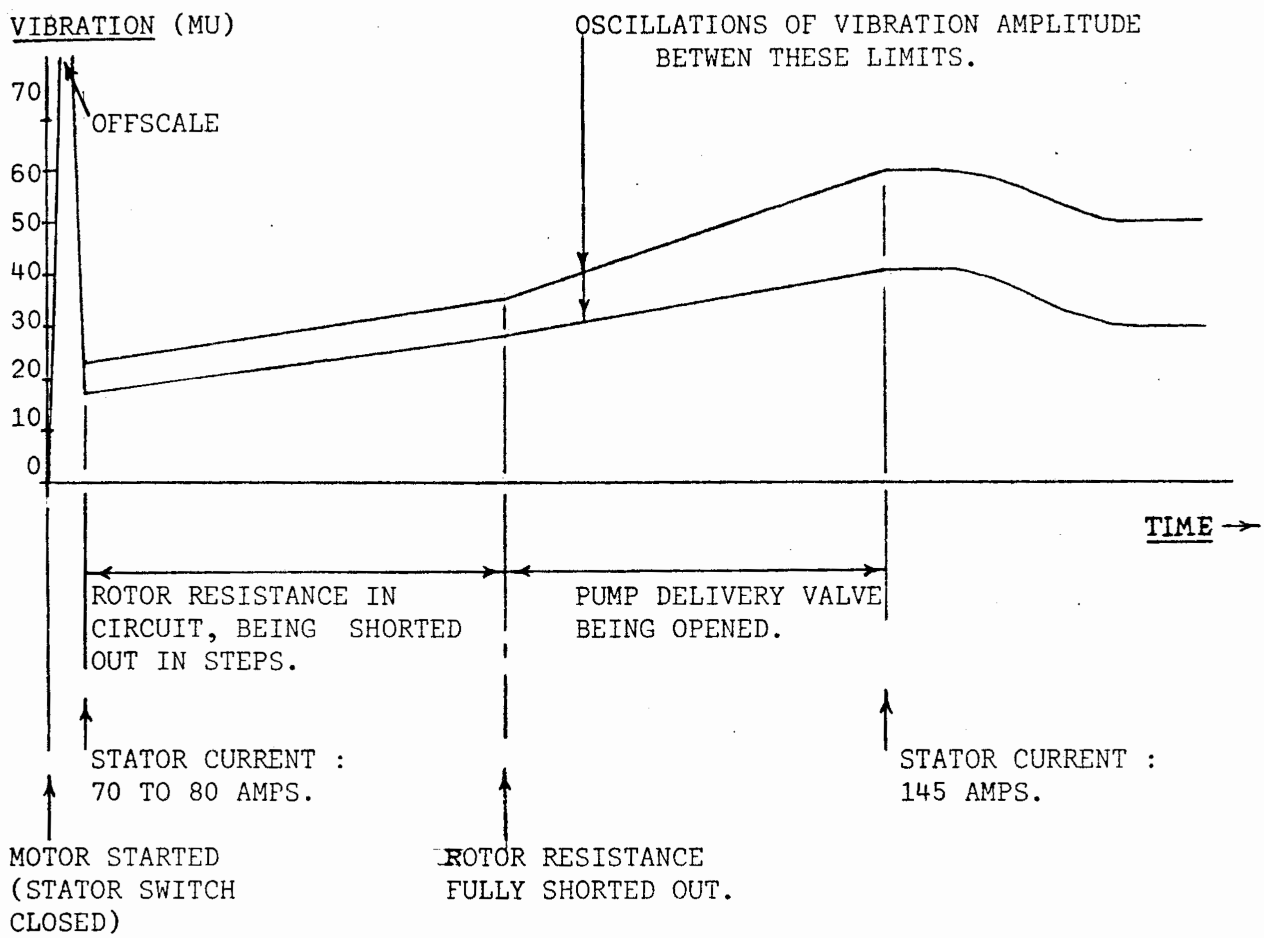
A very rough graphical representation of what happened is given below:-



VIBRATION READINGS DURING STARTING SEQUENCE:

After the "stopping sequence" test the pump was restarted and run for a short period of time to observe what would happen during the starting sequence. These readings were taken on the non-drive-end bearing in the horizontal direction.

A very rough representation of what happened is given below:-



It will be noticed that after the delivery valve is fully open the vibration settles down to a lower value in a few seconds. The oscillations of vibration amplitude appeared to be greater than normal under these conditions but there was no means of checking or repeating the observation. There is a possibility that the vibration would settle down to a different value after a longer period of time and that the oscillations of vibration amplitude may become smaller. If it is considered important enough it should be possible in the future to obtain a more accurate picture of what happens by the use of better instrumentation, particularly by the simultaneous recording of the different quantities.

However, at this stage the vibration readings taken during the stopping and starting sequences demonstrate clearly that the vibration is dependent on the stator current, that is, the vibration is not a purely mechanical vibration. This gives reason to believe that the vibration is associated with the electrical excitation of the motor and therefore that the vibration is associated with unbalanced magnetic pull. As stated in G. RAI's Ph.D. Thesis "The effect of airgap eccentricity on an induction motor" (The City University, London) page 95 "-------vibrations in rotating (electrical) machines are mostly caused by magnetic attraction between various parts of the machine".

GENERAL:

This motor, No---82 had been installed only a few days before the test. It had been overhauled before installation. The test results show that the vibration is much lower than readings taken on other motors of the same set and so the motor can be considered to be in comparatively good condition for this set of motors. Experience has been that a motor if left in operation for some weeks or months will deteriorate in condition from a state corresponding roughly to that of Motor No---82, just tested, to a state in which the horizontal vibration on the bearings will increase to 500, 1000 or more microns peak to peak.

Although the condition of Motor No---82 is comparatively good for this set of motors the vibration on the non-drive-end bearing is above the limit set in BS 2613 : 1970. On this basis alone the performance of the motor is unsatisfactory.

The investigations of the writer in recent years on the reliability of motors has led him to the preparation on behalf of Union Corporation Limited of Enquiry Documents incorporating specifications and criteria not covered by the British Standard Specifications or other specifications. The results of the tests done on those motors which have been designed and manufactured in accordance with Union Corporation's specifications show that the vibration readings taken on these motors have been much lower than the limits laid down in BS 2613 : 1970, that is, always less than 5 per cent of the BS 2613 : 1970 limit in the horizontal direction on the bearings in the factory test. Tests done on such motors as have been commissioned have resulted in readings of vibration in agreement with factory tests or only marginally higher in the coupled-up on-load condition on site. Certain manufacturers have already contracted themselves to Union Corporation to guarantee that the result of the vibration test in the factory will be less than approximately 25 per cent of the BS limit, knowing that the actual results of the vibration tests will be much lower. In the light of these considerations the BS limits for vibration must be considered very liberal. This makes the vibration readings taken on Motor No---82 even more unsatisfactory.

In addition to the vibration on the bearings the higher vibration readings taken on the frame of the motor give cause for concern.

1800 KW, 4 POLE, 6600 VOLT MOTORS FOR MAIN UNDERGROUND PUMPS AT LOWER PUMP STATION AND INTERMEDIATE PUMP STATION, NO. 8 SHAFT, ST. HELENA GOLD MINES LIMITED.

NUMBER OF SLOTS:

The number of stator and rotor slots were counted on a motor that had been dismantled for repairs at Messrs. Reid & Mitchell's workshop, Johannesburg on 9/2/77.

Stator Slots, $S = 72$
Rotor Slots, $R = 60$

$$|S - R| = 72 - 60 = 12$$

According to M. BRADFORD (E.R.A. Report No. 5216, page 9, where he refers to P. VON KAEHNE's E.R.A. Report Ref. Z/T142) :-

$|S - R| \neq 1$ or $2P \pm 1$ to avoid u.m.p. arising from slot combinations. Where P = number of pairs of poles.

(NOTE: P. VON KAEHNE's E.R.A. Report Ref. 2/T142 shows this on page 16).

Now this motor is a 4 pole machine. Therefore P = 2.

So according to this rule -

$|S - R| \neq 1$ This is in order.

or

$$|S - R| \neq 2P \pm 1$$
$$\neq (2 \times 2) \pm 1$$
$$\neq 4 \pm 1$$
$$\neq 3 \text{ or } 5$$

This is in order.

Therefore u.m.p. cannot arise from slot combinations in this motor.

1800 KW, 4 POLE, 6600 V SLIPRING INDUCTION MOTOR DRIVING CENTRIFUGAL PUMP.

TESTS CONDUCTED AT LOWER PUMP STATION, UNDERGROUND NO. 8 SHAFT, ST. HELENA GOLD MINES LIMITED ON MOTOR NO---82 INSTALLED IN NO. 1 POSITION (NEAREST ENTRANCE TO PUMP STATION).

DATE OF TESTS: 2/3/77.

INSTRUMENT USED: IRD MECHANALYSIS PORTABLE VIBRATION METER MODEL NO 306M SERIAL NO. BO 8013641.

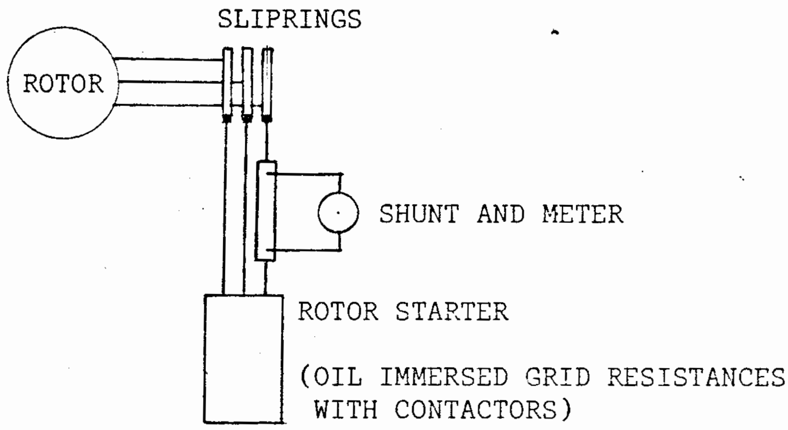
PLEASE NOTE ABBREVIATION: MU = μ = VIBRATION DISPLACEMENT READING, MICRONS PEAK TO PEAK.

IDEAS BEHIND THESE TESTS:

The tests conducted on 22/2/77 demonstrated that there was a regular pulsation of the needle of the vibration meter when the vibration readings were taken on the bearing caps. These pulsations of vibration had been noticed soon after the first of these motors had been commissioned more than a year ago. The motors had vibrated excessively from the start and vibration readings were taken at an early stage. Although they were noticed the pulsations of vibration were not considered significant. However, the studies and investigations on unbalanced magnetic pull which had been made in the meanwhile had brought to hand G. RAI's Ph.D Thesis "The effect of airgap eccentricity on an induction motor" (The City University, London). Considering the vibratory forces that arise from dynamic eccentricity, he states on page 103 "The main vibratory force occurs at rotational speed but the sinusoidal airgap flux density rotates at synchronous speed and as a result the vibratory forces are modulated at twice slip frequency". This theme is elaborated in his work. He also states on page 120 in the same connection that "The spectrum of vibratory forces mainly consisted of a component at rotational speed ...... vibratory forces at twice rotational speed ...... also occurred but their magnitude was quite small". This frequency analysis of the vibration arising from dynamic airgap eccentricity is in complete agreement with the situation found by a frequency analysis done earlier on the pump motors at St.Helena. RAI has also mentioned "beating at twice slip frequency" in connection with static airgap eccentricity in two pole motors on page 128 of his thesis. The presence of the pulsations of vibration in the pump motors at St. Helena therefore became most interesting and this stimulated the tests done on 22/2/77 when the regularity of the pulsations of vibration was checked and the number of pulsations per minute was counted. It will be noticed that the vibration was measured in different directions on the bearing caps. It was found that these were 48 pulsations in the horizontal direction and 24 pulsations in the vertical direction. The pulsations in the axial direction were too erratic to clearly state their frequency. 48 pulsations per minute could possibly correspond to twice slip frequency. This gave rise to certain questions. First, were these pulsations of the vibration in fact related to the slip frequency? Second, if there were 48 pulsations per minute in the horizontal direction and 24 pulsations per minute in the vertical direction, at what angles did 48 p.p.m. change to 24 p.p.m. and then 24 p.p.m. back again to 48 p.p.m. and, in fact, what was the angular distribution of pulsation. It was decided to try to answer these two questions by further tests. Third, why were there 24 p.p.m. at all? If 48 p.p.m. corresponded to twice slip frequency, then 24 p.p.m. corresponded to slip frequency itself. The occurrence of a pulsation of vibration at slip frequency does not appear to be reported in the literature. The answer to the third question is not evident, that is, it is not known at this stage why a pulsation of vibration at slip frequency should occur.

EXPERIMENTAL DETERMINATION OF SLIP FREQUENCY:

A shunt designed for direct current use and rated 3000 amps at 150 millivolts was connected in one phase of the rotor circuit. This was done by breaking into one of the connections of the rotor starter from the motor sliprings as shown in the diagram below:-



The meter used across the shunt was a centre zero milliammeter, because a millivoltmeter was not available.

The milliammeter needle swung regularly to either side of the zero position. The oscillations were very clear and easy to count. The oscillations were counted from a maximum on the positive side, through zero to maximum negative, back through zero to maximum positive again, this counting as one oscillation or pulsation.

At the start of the test 24 pulsations (cycles) were counted over a period of one minute.

During the test 25.5 pulsations were counted over a period of one minute.

At the end of the test 47.5 pulsations were counted over a period of two minutes, that is 23.75 pulsations (cycles) per minute.

VIBRATION READINGS DURING STARTING SEQUENCE:

The pattern of vibration was similar to that recorded in the tests conducted on 22/2/77.

The vibration meter needle went offscale on closing the motor starting switch.

The vibration increased as the speed of the motor increased until the vibration reached a value of 45 to 50 MU at motor full speed but with pump delivery valve closed.

As the valve opened the vibration increased to a value of 48 to 58 MU, changing to a value of 55 to 62 MU and then to a value of 50 to 60 MU.

CONDITION DURING TESTS:

Generally the same as for the tests conducted on 22/2/77. Please refer.

VIBRATION READINGS ON BEARING CAPS (STATOR CURRENT : 144 AMPS):

NON-DRIVE-END BEARING:

ANGLE (DEGREES, REFER DIAGRAM)	RADIAL VIBRATION READING (MU)		PULSATION OF VIBRATION (PULSATIONS PER MINUTE)
	MINIMUM	MAXIMUM	
0	68	76	47
22.5	66	74	47
45	-	-	-
67.5	24	26	47
78.75	68	78	PULSATION ERRATIC. THIS IS ABOUT THE ANGLE AT WHICH PULSATION CHANGES FROM 26 TO 47.
90	20	24	25, 26
112.5	32	34	22 DOUBLE PEAKING.
135	-	-	-
157.5	56	62	23, DOUBLE PEAKING.
168.75	73	80	24, DOUBLE PEAKING.
180	50	62	47
180	68	72	48
191.25	58	64	24, DOUBLE PEAKING.
202.5	56	60	24, DOUBLE PEAKING.
225	-	-	-
247.5	27	31	46, 47.
270	23	25	22, 23, SLIGHTLY ERRATIC, POSSIBLE DOUBLE PEAKING.
292.5	39	42	47
315	-	-	-
337.5	69	78	47

The angular positions were determined from the positions of the eight screw heads on the bearing cap at 22.5, 67.5, 112.5, 157.5, 202.5, 247.5, 292.5 and 337.5 degrees respectively.

The readings at 78.75, 168.75, 180 (second reading) and 191.25 degrees were taken after all the other readings were completed. It is believed that the increased value of vibration given by these readings is due to the increase in temperature of the motor. There was a suggestion made earlier in the investigations on these motors that motor temperature could influence vibration. Some earlier test results also tended to confirm this suggestion. Other tests in which the motor was not actually loaded but in which the temperature was artifically increased by preheating the motor intake air and by restricting the ventilation air quantity proved inconclusive on this point.

The pulsations of vibration were counted as explained in the report on the tests conducted on 22/2/77. It was not always possible to count these accurately because of the limited running time of the pump. The number of pulsations was counted in each case over a period of one minute. The possible error could have been about ± 1 pulsation. It was decided to take as many readings as possible in the limited running time available rather than to strive to obtain higher accuracy in fewer readings. The total time spent on the actual test was about 1½ hours.

In some cases where about 24 pulsations per minute (slip frequency) was counted the needle of the vibration meter hesitated on its way up to the final maximum. This could have been due to a subsidiary peak, as shown in the diagrams. For want of a better term this has been referred to as "double peaking". If the subsidiary peak had been reckoned in, the number of pulsations would have been double the figure given, therefore 48 instead of 24, that is, double slip frequency instead of slip frequency. Therefore in some cases where the vibration pulsates at slip frequency there is an element of pulsation at double slip frequency.

In all cases these readings taken on the bearing caps were radial readings of vibration, that is the probe of the vibration meter was directed to the centre of the bearing cap.

DRIVE-END BEARING:

ANGLE (DEGREES, REFER DIAGRAM)	RADIAL VIBRATION READING (MU)		PULSATION OF VIBRATION (PULSATIONS PER MINUTE)
	MINIMUM	MAXIMUM	
0	21	25	23, VERY ERRATIC.
22.5	27	31	24, DOUBLE PEAKING.
67.5	9	12.5	23.5
90	7	10	24
112.5	7	8	24
157.5	8.5	9.5	24
180	30	35	24, DOUBLE PEAKING.
202.5	4	5.5	24
247.5	2.2	3	24
270	14	17	23 DOUBLE PEAKING.
292.5	7	7.5	23
337.5	8	8.5	23.5 SLIGHT DOUBLE PEAKING.

The methods of measurement on the DE bearing were the same as those described above on the NDE bearing.

CORRELATION BETWEEN PULSATIONS OF VIBRATION AND SLIP FREQUENCY:

These tests have proved a correlation between the pulsations of vibration and slip frequency, that is, measurements in certain positions demonstrate pulsations at twice slip frequency as stated by Rai for dynamic eccentricity. Other measurements demonstrate pulsations of slip frequency itself which, as stated before, does not appear to be reported in the literature and the explanation of which is, at this stage, not known.

In certain cases where pulsations at slip frequency are measured there is an element of pulsation at double slip frequency. This too does not seem to be reported in the literature and the explanation of this also is, at this stage, not known.

THEORETICAL DETERMINATION OF SLIP FREQUENCY:

The slip frequency has been determined experimentally as already explained.

The slip frequency will now be determined theoretically from fundamentals and from the rating plate data of the motor.

$$\frac{f_2}{f} = \frac{n_s - n}{n_s} = \frac{\frac{f}{p} - n}{\frac{f}{p}}$$

where, f_2 = slip frequency
f = frequency of supply
= 50 Hertz (cycles per second)

p = number of pole-pairs
= 2 (4 pole motor)

$$\frac{f_2}{f} = \frac{f - pn}{f}$$

n_s = synchronous speed
$= \frac{f}{p} = \frac{50}{2}$ = 25 revs per second
= 25 x 60 = 1500 revs per minute.

$$f_2 = f - pn$$

n = rated full load speed
= 1485 r.p.m.
$= \frac{1485}{60}$ revs per second

slip frequency, $f_2 = (50 - \frac{2.1485}{60})$ cycles per second.

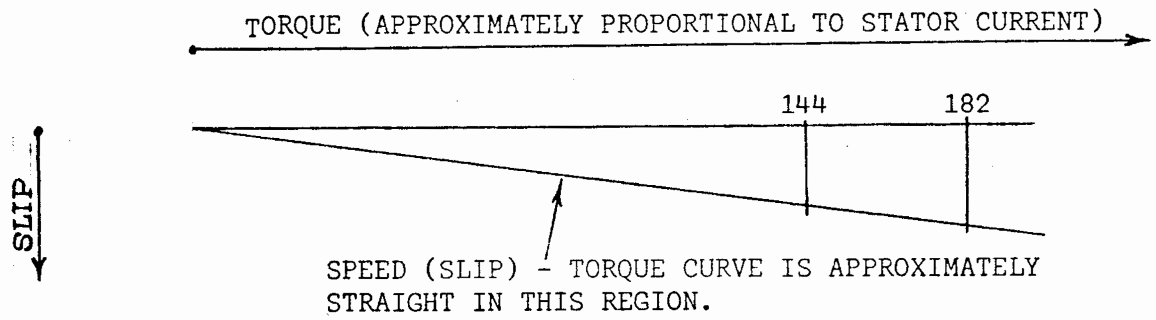
$= (50 - \frac{2.1485}{60}) \times 60$ cycles per minute.

= (3000 - 2.1485) c.p.m.

= (3000 - 2970) c.p.m.

= 30 c.p.m.

This is at full load rated current = 182 amps. but at actual current during tests of 144 amps,



Therefore actual slip frequency at a load of 144 amps

$$= \quad 30 \times \frac{144}{182}$$

$$= \quad 23.74 \text{ cycles per minute.}$$

This agrees well with the experimental determination of slip frequency.

CONSISTENCY OF RESULTS:

The results of the tests conducted on 2/3/77 are not entirely consistent with the results of the tests conducted on 22/2/77. The levels of vibration are different.

Also the horizontal pulsations of vibration on the drive end bearing were found to be 48 pulsations per minute on 22/2/77, but 24 pulsations per minute, double peaking on 2/3/77. The results are nevertheless reported as found.

Although previous investigations on these motors were not conducted in such detail it became clear that the level of vibration gradually increased over some weeks or months. Moving a motor from one position to another resulted in a change in vibration. Minor adjustments to motor mountings or to the motor alignment resulted in a change in vibration. For these reasons the motors have come to be regarded as unstable with respect to their vibration. A degree of inconsistency in the results over a week or more is to be expected in these circumstances.

1800 KW, 4 POLE, 6600 V SLIPRING INDUCTION MOTOR DRIVING CENTRIFUGAL PUMP. TESTS CONDUCTED AT INTERMEDIATE PUMP STATION, UNDERGROUND, NO. 8 SHAFT, ST. HELENA GOLD MINES LIMITED ON MOTOR NO---78 INSTALLED IN NO. 1 POSITION (NEAREST ENTRANCE TO PUMP STATION).

DATE OF TESTS: 8/3/77.

INSTRUMENT USED: IRD MECHANALYSIS PORTABLE VIBRATION METER MODEL NO. 306M SERIAL NO. BO 8013641.

PLEASE NOTE ABBREVIATION : MU= μ = VIBRATION DISPLACEMENT READING, MICRONS PEAK TO PEAK.

BACKGROUND:

The tests conducted on this motor were a continuation of the latest series of tests on this set of motors, particularly the tests dated 22/2/77 and 2/3/77. The pumping period at the Intermediate Pump Station tends to be much shorter than at the Lower Pump Station. It was therefore necessary to limit the number of readings taken. It was decided to take as many readings as possible on the motor frame, bedplate and foundation and to restrict the number of readings of vibration on the bearing caps.

VIBRATION READINGS DURING STARTING SEQUENCE:

Readings of vibration were taken on the non-drive-end bearing cap in the horizontal direction. The pattern of vibration was similar to that recorded in the tests conducted on 22/2/77.

The vibration meter needle went offscale on closing the motor starting switch followed by a rapid drop to a low value and then a gradual increase from about 20 MU to a value of about 70 to 80 MU.

EXPERIMENTAL DETERMINATION OF SLIP FREQUENCY:

The slip frequency was determined experimentally as described in the report of the tests conducted on 2/3/77.

At the start of the tests 23.5 pulsations or cycles of rotor current were counted over a period of one minute. Towards the end of the tests 22 pulsations of rotor current were counted over a period of one minute.

STATOR CURRENT:

The stator current was 142 amps. It was noticed that the ammeter needle was not absolutely steady but was pulsating very slightly. It was decided to count these pulsations. 44 pulsations per minute of stator current were counted at the time the frequency of the rotor current was 22 cycles per minute, that is, stator current was pulsating at twice slip frequency. Putting it another way, the stator current was amplitude modulated at twice slip frequency.

The distribution switchgear supplying the pumps at the Lower Pump Station is installed at the Intermediate Pump Station. It was therefore possible to use the ammeters on this switchgear to observe the stator current of the motor that was operating at that time at the Lower Pump Station. The amplitude modulation of the stator current was shown even more clearly on these ammeters. This was applicable to Motor No---79 installed in the No.2 (middle) position at the Lower Pump Station, so details of these readings are included in the reports of the tests conducted on 8/3/77 on Motor No---79.

The literature does not appear to make any mention of the amplitude modulation of stator current at twice slip frequency or of its association with the pulsation or amplitude modulation of vibration at twice slip frequency or at slip frequency itself and of the association this has with the operation of the motor under conditions of airgap eccentricity, particularly of dynamic eccentricity. This behaviour of the stator current may serve as a means of gaining an insight into the conditions within the motor while the motor is operating, particularly conditions of airgap eccentricity giving rise to unbalanced magnetic pull and vibratory forces. The observations made on 8/3/77 regarding stator current modulation have been on fairly well loaded motors having significant slip. Observations under various conditions of load and slip, in cases where it is possible to vary the load, may give a better understanding of the effect.

VIBRATION READINGS ON BEARING CAPS (STATOR CURRENT 142 AMPS):

NON-DRIVE-END BEARING:

ANGLE (DEGREES, REFER DIAGRAM)	RADIAL VIBRATION READING (MU)		PULSATION OF VIBRATION (PULSATIONS PER MINUTE)
	MINIMUM	MAXIMUM	
0	88	100	48
90	3	5	23
180	68	78	45
270	17	21	47

DRIVE-END BEARING:

ANGLE (DEGREES, REFER DIAGRAM)	RADIAL VIBRATION READING (MU)		PULSATION OF VIBRATION (PULSATIONS PER MINUTE)
	MINIMUM	MAXIMUM	
0	76	84	47
90	30	32	47
180	74	82	47
270	29	31	47

VIBRATION READINGS ON MOTOR FRAME, BEDPLATE AND FOUNDATION:
(STATOR CURRENT 142 AMPS)

POSITION	HORIZONTAL VIBRATION READING (MU)			
	DRIVE-END		NON-DRIVE-END	
	MINIMUM	MAXIMUM	MINIMUM	MAXIMUM
A	0.5	0.5	0.5	1
B	1	1.5	1	3
C	5	5.5	5	6
D	7.5	8.5	5	6
E	16	18	14	15
F1	24	26	23	25
F2	40	42	38	40
G	52	58	56	62
H	54	60	62	68
I	68	78	82	88
J	82	98	95	105

Please refer diagram showing positions where readings were taken and graphical plot of readings.

The results are generally similar to the results of the same tests conducted on 22/2/77. The comments on the tests done on 22/2/77 therefore apply also to these latest tests. In this latest test two readings were taken near the position previously designated F where most of the cracks have occurred on the motor frame just above the gussets joining the feet of the motor to the barrel of the motor frame. These two readings were done at positions designated F1 and F2. The graphical plot of the readings shows the sharp increase in vibration between positions F1 and F2. The level of the vibration in the foundation, bedplate and foot of motor is low for the tests on both motors tested, but the level of vibration higher on the motor frame, whether on the drive-end or on the non-drive-end, increases in a manner which is not related to the level of vibration in the foundation, bedplate or foot or even to the level of vibration on the motor frame below position F. This can be seen by the crossing of the drive-end graph and non-drive-end graph of each motor. Also if the graphs of both motors are superimposed by plotting the graphs on tracing paper it can be seen that for the latest tests (8/3/77, Motor No---78) the vibration has a higher value low on the motor frame but that the vibration has a lower value high on the motor frame as compared to the results of the earlier tests (22/2/77, Motor No---82). That is the graphs of the two motors cross each other. The way in which the vibration increases as one progresses higher up on the motor is a characteristic of the motor concerned and not of the foundation and bedplate. Also, if the bedplate at the Intermediate Pump Station is more effectively grouted in than the bedplate at the Lower Pump Station to an extent which can cause (!) the vibration, as claimed by the motor manufacturer, or even if this be to an extent which can permit excessive vibration, then one would expect less vibration on the IPS motor. The facts are that there is significantly more vibration on the IPS motor than on the LPS motor at positions lower on the motor frame and on the bearings generally but especially on the drive-end bearing.

1800 KW, 4 POLE, 6600 V SLIPRING INDUCTION MOTOR DRIVING CENTRIFUGAL PUMP.

TESTS CONDUCTED AT LOWER PUMP STATION, UNDERGROUND, NO. 8 SHAFT, ST. HELENA GOLD MINES LIMITED ON MOTOR NO---79 INSTALLED IN NO. 2 POSITION (MIDDLE POSITION).

DATE OF TESTS: 8/3/77.

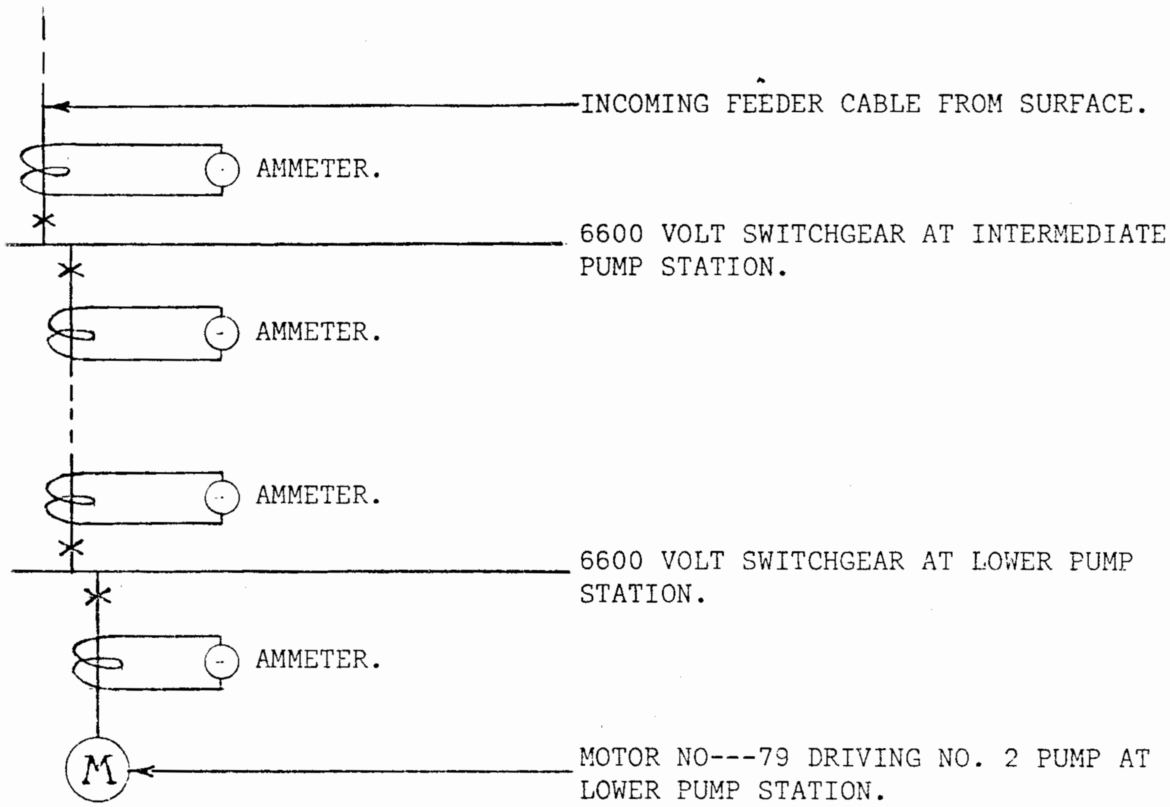
INSTRUMENT USED: IRD MECHANALYSIS PORTABLE VIBRATION METER MODEL NO. 306 M SERIAL NO. BO 8013641.

PLEASE NOTE ABBREVIATION: MU = μ = VIBRATION DISPLACEMENT READING, MICRONS PEAK TO PEAK.

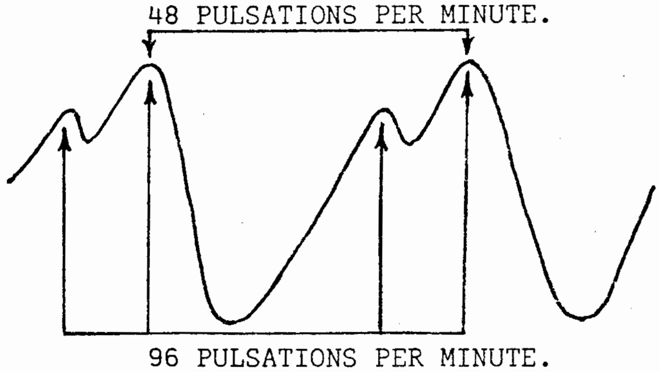
STATOR CURRENT OBSERVATIONS AT INTERMEDIATE PUMP STATION:

Before reporting the tests done on Motor No---79 at the Lower Pump Station certain observations made at the Intermediate Pump Station will be described. This was referred to earlier in this report in the paragraph headed "STATOR CURRENT".

The distribution switchgear arrangement is given in the diagram below.



From this diagram it can be seen that the stator current of the motor operating at the Lower Pump Station can be read on the indicating ammeters mounted on the switchgear at the Intermediate Pump Station. The reading on both these ammeters was 150 amps but the needle of the ammeter was oscillating at exactly 48 pulsations per minute with a subsidiary oscillation giving a "double peaking" effect, somewhat as shown in the diagram below.



If the double peaking effect were taken into account there would be 2 x (the number of counted pulsations per minute), that is 2 x 48 p.p.m. = 96 p.p.m.

On the basis of the tests which had just been done on Motor No---78 at IPS one would expect 48 p.p.m. to be twice slip frequency so that slip frequency would be 24 p.p.m.

Motor No---79 at LPS had recently been completely overhauled and rebalanced. It had been installed to drive No. 2 Pump on 5/3/77 that is only three days before this test.

VIBRATION READINGS ON BEARING CAPS (STATOR CURRENT 145 AMPS):

These tests were done on Motor No---79 in No. 2 Position at LPS after completing the above described stator current observations at IPS. The stator current on Motor No---79 in No. 2 Position at LPS was read on the indicating ammeter on the instrumentation panel at LPS.

NON-DRIVE-END BEARING:

ANGLE (DEGREES, REFER DIAGRAM)	RADIAL VIBRATION READING (MU)		PULSATION OF VIBRATION (PULSATIONS PER MINUTE)
	MINIMUM	MAXIMUM	
0	145	155	-
22.5	160	170	-
45	145	150	-
67.5	115	120	-
78.75	100	110	21 DOUBLE PEAKING
90	69	72	21
101.25	33	36	-
112.5	28	32	-
135	58	78	-
157.5	125	135	42
180	140	150	42
202.5	160	170	-
225	138	145	-
247.5	120	125	-
270	50	52	-
292.5	22	24	-
315	62	70	-
337.5	108	115	-

The polar diagram of these vibration readings resembles two lobes on a straight axis oriented at 22.5° (202.5°). The maximum readings of vibration at 22.5° and 202.5° respectively are equal to one another. This suggested that the lobes may be circles. To investigate this a tracing was made of the polar diagram of these vibration readings except that two circles are superimposed upon it. It can be seen that the graphical plots of the vibration readings lie approximately on these two circles.

These two circles would therefore have the following properties:-

1. The two circles touch each other at the centre of the polar diagram of vibration readings.

2. The common tangent to the two circles is oriented at 112.5° and 292.5°.

3. The two circles each have a diameter normal to the common tangent. These diameters are oriented at 22.5° and 202.5° respectively. These two diameters form a straight line through the centre of the polar diagram of vibration readings.

4. The vibration readings would ideally be concyclic with these two circles.

It will be seen that the maximum value of vibration is not necessarily in the horizontal direction. British Standard BS 2613 : 1970 calls for vibration readings in the horizontal direction only when testing motors.

If the vibration readings are a maximum in a particular direction it would suggest that the vibratory forces causing the vibration would be a maximum in the same direction. Therefore, to the extent that these vibratory forces are associated with unbalanced magnetic pull (u.m.p.), the u.m.p. itself would be a maximum in this direction.

DRIVE-END BEARING:

ANGLE (DEGREES, REFER DIAGRAM)	RADIAL VIBRATION READING (MU)		PULSATION OF VIBRATION (PULSATIONS PER MINUTE)
	MINIMUM	MAXIMUM	
0	140	160	-
90	90	96	-
180	140	160	-
270	74	78	-

Readings of radial vibration have been plotted graphically. Because the pulsations of vibration were not counted in this case, a diagram for pulsation of vibration has not been drawn.

VIBRATION READINGS DURING STOPPING SEQUENCE:

Readings of vibration were taken on the non-drive-end bearing cap in the horizontal direction. The pattern of vibration was similar to that observed in the tests conducted on 22/2/77.

The pump delivery valve was gradually closed. During this period the vibration decreased progressively from a value of 140 to 160 MU (oscillating) to a value of 120 to 130 MU (oscillating). At this point the operators tripped the motor. Immediately the motor tripped the vibration dropped to zero, then oscillated up and down twice and then decayed to zero. Some time after this there was an increase followed by a decrease in vibration as the motor slowed down, possibly due to passing through a critical speed.

STATOR CURRENT OBSERVATIONS AT LOWER PUMP STATION:

While the motor was driving the pump the current registered on the indicating ammeter on the instrumentation panel at LPS was 145 amps. The ammeter was not absolutely steady but was pulsating very slightly. 41 pulsations of the ammeter needle were counted over a period of one minute. A second count over one minute gave 42 pulsations. These pulsations were double peaking so if this is taken into account there would be 82 or 84 pulsations per minute.

On the basis of the tests which had just been done on Motor No---78 at IPS, one would expect 41 or 42 p.p.m. to be twice slip frequency so that slip frequency would be 20.5 or 21 p.p.m.

As can be seen the vibration pulsated at 21 p.p.m. and 42 p.p.m. which should correspond to slip frequency and twice slip frequency respectively from previous tests.

However the stator current pulsations counted at the Intermediate Pump Station were 48 p.p.m. (96 p.p.m. if double peaking taken into account). Possibly the slip frequency itself changed between the observations at IPS and the observations at LPS.

1800 KW, 4 POLE, 6600 V SLIPRING INDUCTION MOTOR

DRIVING CENTRIFUGAL PUMP.

TESTS CONDUCTED AT LOWER PUMP STATION, UNDERGROUND, NO. 8 SHAFT,

ST. HELENA GOLD MINES LIMITED

ON MOTOR NO---82 INSTALLED IN NO. 1 POSITION

(NEAREST ENTRANCE TO PUMP STATION.)

DATE OF TESTS: 15/3/77.

INSTRUMENT USED: IRD MECHANALYSIS PORTABLE VIBRATION METER
MODEL NO. 306 M SERIAL NO. BO 8013641.

PLEASE NOTE ABBREVIATION: MU = μ = VIBRATION DISPLACEMENT READING, MICRONS PEAK TO PEAK.

VIBRATION READINGS ON BEARING CAPS (STATOR CURRENT 144 AMPS):

NON-DRIVE-END BEARING (EARLIER READINGS, MOTOR COOL):

ANGLE (DEGREES, REFER DIAGRAM)	RADIAL VIBRATION READING (MU)		PULSATION OF VIBRATION (PULSATIONS PER MINUTE)
	MINIMUM	MAXIMUM	
0	26	32	49
22.5	28	38	49
67.5	15	20	49
90	16	17	48
112.5	34	36	48
157.5	15	21	ERRATIC
180	34	42	47
202.5	13	15	24 DOUBLE PEAKING
247.5	7	8	24
270	10	11	ERRATIC
292.5	18	22	49
337.5	28	32	50

NON-DRIVE-END BEARING (LATER READINGS, MOTOR WARMER):

ANGLE (DEGREES, REFER DIAGRAM)	RADIAL VIBRATION READING (MU)		PULSATIONS OF VIBRATION (PULSATIONS PER MINUTE)
	MINIMUM	MAXIMUM	
0	56	64	50
22.5	50	60	49 VERY DISTINCT
67.5	18	22	47
90	18	20	24 DOUBLE PEAKING
112.5	28	32	25 DOUBLE PEAKING
157.5	54	60	24 DOUBLE PEAKING
180	50	56	49
202.5	44	50	25 VERY DISTINCT DOUBLE PEAKING
247.5	17	20	24 VERY DISTINCT DOUBLE PEAKING
270	18	20	ERRATIC
292.5	26	30	25 DOUBLE PEAKING
337.5	56	64	50

DRIVE-END BEARING:

ANGLE (DEGREES, REFER DIAGRAM)	RADIAL VIBRATION READING (MU)		PULSATIONS OF VIBRATION (PULSATIONS PER MINUTE)
	MINIMUM	MAXIMUM	
0	24	28	NOT COUNTED
90	18	23	NOT COUNTED
180	18	22	NOT COUNTED
270	22	24	NOT COUNTED

VIBRATION READINGS ON MOTOR FRAME (STATOR CURRENT : 144 AMPS):

POSITION	HORIZONTAL VIBRATION READING (MU)			
	TOP EDGE		AXIS	
	MINIMUM	MAXIMUM	MINIMUM	MAXIMUM
XA	-	-	22	25
XB	50	60	28	30
XC	180	200	96	100
XD	200	210	100	104
XE	115	118	-	-
XF	-	-	100	105
XG	105	110	88	92
XH	-	-	74	80
XI	100	105	92	98
XJ	95	100	80	84
XK	-	-	46	50
XL	-	-	50	54

Please refer to diagram showing positions where readings were taken and graphical plot of readings.

The readings were taken in a similar way to the readings taken on 22/2/77. These readings (taken on 15/3/77) should be compared with the readings taken on 22/2/77. It will be seen that the vibration on the top edge of the motor near the spiggot of the drive-end endshield has increased from values of 50 to 70 MU to 180 to 220 MU, that is, an increase of more than 3 to 1. The reason for this remarkable increase in vibration is not known. There has in addition been marginal increase in vibration at other positions along the top edge of the frame. As can be seen all the vibration readings fluctuate between minimum and maximum values, that is, the vibration pulsates, or in other words, the vibration is amplitude modulated. This amplitude modulation of the vibration is common to the frame and the bearings. The level of vibration is higher on the axis of the frame than on the bearings except for one reading of vibration on the non-drive-end endshield. The level of vibration on the top edge of the motor is higher than the vibration on the axis of the motor and is higher than the vibration on the bearings.

VERTICAL VIBRATION READINGS ON MOTOR FRAME:

Vertical vibration readings were taken near position XJ on the top edge of the motor frame under the same conditions as for the horizontal vibration readings recorded above. These vertical vibration readings were taken just after the horizontal vibration readings were taken.

The vertical vibration reading on the frame itself near position XJ was only 11 to 14 MU.

The vertical vibration reading on the endshield near position XJ was only 15 to 17 MU.

As already recorded, the horizontal vibration at this point was 95 to 100 MU, that is, roughly 6 or more times the vertical vibration.

VIBRATION READINGS DURING STARTING SEQUENCE:

Readings of vibration were taken on the non-drive-end bearing in the horizontal direction during the starting sequence. The pattern of vibration was similar to that recorded in the tests conducted on 22/2/77 and 8/3/77. The vibration meter needle went offscale on closing the starting switch followed by a rapid drop to a low value and then a gradual increase from about 15 MU to a value of 45 to 55 MU, amplitude modulated at 47 pulsations per minute.

VIBRATION READINGS DURING STOPPING SEQUENCE:

Readings of vibration were taken on the non-drive-end bearing cap in the horizontal direction during the stopping sequence. The pattern of vibration was similar to that observed in the tests conducted on 22/2/77 and 8/3/77. The pump delivery valve was gradually closed. During this period the vibration decreased gradually from a value of 54 to 58 MU (oscillating) to roughly 50 MU (oscillating). At this point the operators tripped the motor. Immediately the motor tripped the vibration dropped to zero, then oscillated up and down twice and then decayed to zero. On this occasion no further note was taken of any possible vibration as the motor slowed down.

VERTICAL VIBRATION READINGS ON MOTOR FRAME:

Vertical vibration readings were taken near position XJ on the top edge of the motor frame under the same conditions as for the horizontal vibration readings recorded above. These vertical vibration readings were taken just after the horizontal vibration readings were taken.

The vertical vibration reading on the frame itself near position XJ was only 11 to 14 MU.

The vertical vibration reading on the endshield near position XJ was only 15 to 17 MU.

As already recorded, the horizontal vibration at this point was 95 to 100 MU, that is, roughly 6 or more times the vertical vibration.

VIBRATION READINGS DURING STARTING SEQUENCE:

Readings of vibration were taken on the non-drive-end bearing in the horizontal direction during the starting sequence. The pattern of vibration was similar to that recorded in the tests conducted on 22/2/77 and 8/3/77. The vibration meter needle went offscale on closing the starting switch followed by a rapid drop to a low value and then a gradual increase from about 15 MU to a value of 45 to 55 MU, amplitude modulated at 47 pulsations per minute.

VIBRATION READINGS DURING STOPPING SEQUENCE:

Readings of vibration were taken on the non-drive-end bearing cap in the horizontal direction during the stopping sequence. The pattern of vibration was similar to that observed in the tests conducted on 22/2/77 and 8/3/77. The pump delivery valve was gradually closed. During this period the vibration decreased gradually from a value of 54 to 58 MU (oscillating) to roughly 50 MU (oscillating). At this point the operators tripped the motor. Immediately the motor tripped the vibration dropped to zero, then oscillated up and down twice and then decayed to zero. On this occasion no further note was taken of any possible vibration as the motor slowed down.

INFLUENCE OF TEMPERATURE ON VIBRATION:

The following is a speculation on the readings of vibration on the non-drive-end bearing:-

1) The level of vibration for readings taken later is higher than the level of vibration for readings taken earlier. This applies for readings in almost all directions. This increase in vibration level over a period of time is believed to be due to the increase in temperature of the motor as it operates under load during this period of time. This theory has been referred to in the report on the tests conducted on 2/3/77. According to this theory an increase in temperature would cause thermal expansion of the rotor. This thermal expansion would not necessarily be uniform in different parts of the rotor so that the rotor would bend as a result. This would cause an increase in the mechanical dynamic unbalance of the rotor. It is known that mechanical dynamic unbalance gives rise to vibration of frequency equal to the rotational speed of the rotor.

2) As the temperature of the motor increases there seems to be a tendency for the vibration to be amplitude modulated at slip frequency itself (with a double peaking effect) rather than at twice slip frequency, that is, at the higher temperature the results of this test show that amplitude modulation at slip frequency itself (with a double peaking effect) is spread over a bigger arc than is the case at a lower temperature.

The possibility of an interrelation between these two effects is suggested but not proved.

1800 KW, 4 POLE, 6600 V SLIPRING INDUCTION MOTOR DRIVING CENTRIFUGAL PUMP. TESTS CONDUCTED AT LOWER PUMP STATION, UNDERGROUND, NO. 8 SHAFT, ST. HELENA GOLD MINES LIMITED ON MOTOR---82 INSTALLED IN NO. 1 POSITION (NEAREST ENTRANCE TO PUMP STATION).

DATE OF TESTS: 22/3/77.

INSTRUMENTS USED:

(Please refer SCHEMATIC DIAGRAM OF CONNECTIONS).

Chart Recorder:

WATANABE Multirecorder.

The following signals were fed to this chart recorder:-

Rotor Current (Red Pen).
Modulation of Vibration (Purple Pen).
Modulation of Stator Current (Brown Pen).

Vibration Analyser:

IRD MECHANALYSIS Model 340 vibration analyser.
The oscilloscope output of this vibration analyser was fed through the Rectifier and Filter Circuit to the Chart Recorder to give a record of the modulation of vibration.

Rectifier and Filter Circuit:

This circuit was designed and built by MR.T. CORFIELD, UNION CORPORATION GROUP ELECTRONICS LABORATORY.

This circuit processed the oscilloscope output of the vibration analyser. The output of this circuit was fed to the Chart Recorder to give a record of the modulation of vibration.

Stator Current Instrumentation:

The existing current transformer installed in the 6600 volt switchgear and the existing indicating ammeter installed on the main control panel in the pump station were used to measure and observe the stator current and also to provide a signal from the ammeter terminals through the Sample and Hold Circuit to the Chart Recorder to give a record of the modulation of the stator current.

Sample and Hold Circuit:

This circuit was designed and built by MR. T. CORFIELD, UNION CORPORATION GROUP ELECTRONICS LABORATORY.

This circuit processed the signal on the ammeter terminals. The output of this circuit was fed to the Chart Recorder to give a record of the modulation of stator current.

Rotor Current Instrumentation:

A shunt designed for d.c. use, and rated 150 millivolts at 3000 amps, was used with a milliammeter, as described in detail in "Experimental Determination of Slip Frequency" in the report of tests conducted on 2/3/77. The signal from the milliammeter terminals was fed to the Chart Recorder to give a record of rotor current.

Vibration Meter:

IRD MECHANALYSIS PORTABLE VIBRATION METER MODEL NO. 306 M SERIAL NO. BO 8013641.

This vibration meter was used to measure vibration and the pulsations of vibration directly when its probe was placed at various positions on the motor.

PLEASE NOTE ABBREVIATION : MU = μ = VIBRATION DISPLACEMENT READING MICRONS PEAK TO PEAK.

EXPERIMENTAL DETERMINATION OF SLIP FREQUENCY:

The slip frequency was determined experimentally as described in the report of tests conducted on 2/3/77. The motor was running at the start of the tests, having been in use for some time before, so the motor was warm. 24,5 pulsations or cycles of rotor current were counted over a period of one minute. The amplitude of the oscillations of rotor current were rather weaker than had been observed in earlier tests. In earlier tests the needle of the milliammeter connected across the shunt had oscillated approximately 38-0-38 milliamps, but in this test the needle oscillated 16-0-16 milliamps. Possibly this was cased by a low substation voltage of 6,2 kV which was read during the tests, as compared with the rated voltage of 6,6 kV. 6,2 kV represents 93.9% of rated voltage. BS 2613 : 1970 specifies that a motor should be capable of satisfactory operation between 95% and 105% of normal voltage. The voltage is usually between 95% and 100% of normal voltage in this particular installation although there are many installations where the voltage falls below 95% of rated voltage for long periods.

STATOR CURRENT:

The stator current as read on the existing ammeter installed on the main control panel in the pump chamber was 142 amps. The ammeter needle was oscillating very slightly. The frequency of the oscillations of the ammeter needle was so high that the number of oscillations could not be counted. It was then decided to take readings of the stator current on the existing ammeter installed on the 6600 volt switchpanel in the substation near this pump station. Again it was found difficult to count the oscillations but after several attempts it was possible to count roughly 47 main pulsations over a period of one minute. These main pulsations were complicated by double peaking and subsidiary oscillations. A subsequent attempt yielded a count of roughly 45 main pulsations over a period of one minute with the same conditions of complication and difficulty of counting. The pulsation of oscillation was definitely not as clear nor as easy to count as had been the case in previous tests. Some change in conditions must have taken place to account for this difference. Suggested possible changes are the reduced value of the supply voltage and/or lower amplitude of rotor currents discussed in "Experimental Determination of Slip Frequency" above.

The pulsations of stator current, counted as roughly 45 or 47 pulsations per minute, can be taken as twice the rotor current frequency of 24,5 pulsations per minute if one takes into account the inaccuracy caused by the difficulty in counting the stator current pulsations on this occasion. This is the same result as obtained by counting the pulsations of the stator current in the tests conducted on 8/3/77.

The pulsations of stator current were measured in later tests conducted on 22/3/77 using more sophisticated instrumentation, that is, the stator ammeter signal was processed by a Sample and Hold Circuit. The output of this circuit was fed to a Chart Recorder to give a record of the modulation of stator current. These charts form a part of this report and it can be seen from these charts that the modulation of stator current at twice slip frequency is present, though somewhat irregularly, and complicated by the fact that oscillations of still higher harmonics of the slip frequency are present. It would appear that the stator current is not only amplitude modulated but that it is also frequency modulated. If one considers the damping of the ammeter movement then the observations that were made of the oscillations of the ammeter needle make sense. In the case where the ammeter movement is very lightly damped the oscillations of the needle would correspond to the higher harmonics of the modulation and so they would be too fast to count manually. On the other hand in the case where the ammeter is more heavily damped the oscillations of the needle would correspond to the lowest harmonic of the modulation which would be twice slip frequency, which is easy to count manually, with indications of "double peaking" and higher harmonics which cannot be counted manually but which can still be observed.

Further details of the modulations of stator current are given on the recorder charts themselves. These charts have been annotated and are discussed separately.

VIBRATION READING ON NON-DRIVE-END BEARING CAP (STATOR CURRENT 142 AMPS):

ANGLE (DEGREES, REFER DIAGRAM)	RADIAL VIBRATION READING (MU)		PULSATION OF VIBRATION (PULSATIONS PER MINUTE)
	MINIMUM	MAXIMUM	
0	40	48	48,5
90	17	19	ERRATIC PULSATIONS, COULD NOT BE COUNTED
180	42	52	48,5
270	21	23	ERRATIC PULSATIONS, COULD NOT BE COUNTED

COMMENTS ON RECORDER CHARTS:

Detailed notes have been made on each recorder chart. For general conclusions arising from the recorder charts please refer to the section of the report on these tests headed "Conclusions". The following notes are intended to assist in an understanding of the charts.

The 17 charts included in this report were selected from a much longer length of recorder chart made in the course of the tests. This selection was made in order to give a reasonably concise but accurate record of what transpired at the tests.

Charts Nos. 1 to 4. Motor No---82:

As these charts were being recorded the traces appeared to be irregular. It was believed that these results were not consistent with the results of the tests done on earlier dates with less sophisticated test equipment. In particular the earlier observations had shown that the vibration was modulated at slip frequency (sometimes with double peaking) or at twice slip frequency, and that the stator current was modulated at twice slip frequency (with what appeared to be double peaking). These charts, however, produced a much more complicated picture. Also, the observations of stator current on an indicating ammeter at the present tests appeared to be different to earlier observations. It was believed that there had been a change in conditions. Therefore, after taking some observations and charts on Motor No---82 it was decided to stop this motor and to take readings on Motor No---79. As will be seen from Chart No. 4, which was recorded towards the end of the tests on Motor No---82, it was only on the frame of this motor that a reasonably simple modulation of vibration at slip frequency was obtained. This vibration was very high and this simpler picture of modulation was obtained only at reduced amplitude.

Charts Nos. 5 to 17. Motor No---79:

On starting Motor No---79 the modulations of vibration and of stator current appeared to be even more irregular than the modulations on Motor No---82. It was, at first, believed to be due to Motor No---79 being too cool, not having run for some time.

Therefore the motor was run to warm it up. Successive readings were made with little, if any, improvement in the regularity of the vibrations.

At this stage it was decided to take readings with the simple hand held vibration meter. These readings showed clear modulations of vibration at twice slip frequency. At the same time the needle of the indicating meter on the vibration analyser was fluctuating erratically. It was then thought that this was due to the simpler vibration meter having more damping than the indicating meter on the vibration analyser. The chart recordings which had been taken through the vibration analyser had, up to that point in time, been taken with the vibration analyser setting in the "Filter Out" position, that is, the vibration was not filtered. It was decided to change this vibration analyser setting to "Sharp filter". This filter was tuned on the vibration analyser to the rotational speed of the motor. As can be seen on Chart No. 8, this immediately changed the record of the modulation of vibration to a clear modulation at twice slip frequency.

In Chart No. 9 the vibration analyser setting was changed from "Sharp Filter" to "Broad Filter". The filter was tuned to the rotational speed of the motor. This also gave a very clear record of the modulation of vibration at twice slip frequency with a very slight increase in the irregularity of modulation. The vibration analyser was set in this "Broad Filter" position for the remainder of the tests. Observations of vibration were made on various positions on the motor and these observations were recorded on the charts.

Charts Nos. 8 to 10 inclusive show that for horizontal vibration on the non-drive-end bearing cap the vibration is modulated at twice slip frequency. However, Charts Nos. 11 and 12 show that for vertical vibration on the non-drive-end bearing cap the vibration is modulated at slip frequency itself. This is all in accordance with observations made on earlier dates with the hand held vibration meter.

On the drive-end bearing both the horizontal and vertical vibration are shown on Charts Nos. 14 and 15 to be modulated at twice slip frequency. Chart No. 16 shows the axial vibration on the non-drive-end bearing to be modulated very irregularly even with the broad filter in use. These observations also are in accordance with observations made on earlier dates with the hand held vibration meter.

Chart No. 17 was taken during the course of stopping this motor. It can be seen that the stator current decreases as the load is taken off the motor. The vibration decreases very slightly while this takes place. At the instant the motor is tripped there is a simultaneous disappearance of the stator current, the vibration and the rotor current. This too is in accordance with observations made on earlier dates.

CONCLUSIONS:

The relationships between the rotor current (slip frequency), the modulation of vibration and the modulation of stator current will be seen clearly from the recorder charts and the discussions on these recorder charts.

In brief the recorder charts have revealed the following:-

1) The vibration is amplitude modulated at twice slip frequency (main peaks) and simultaneously at 8 times slip frequency (all peaks).

The amplitude modulation of vibration is not absolutely regular. Sometimes a peak is missed when one would expect a peak to appear. This irregularity complicates the main modulation. This irregularity may temporarily disturb the main basic modulation but in the longer term the modulation maintains a basic regularity. This irregularity in the modulation is believed to be a manifestation of frequency modulation in the presence of the basic amplitude modulation.

Ignoring these irregularities, and considering only the main peaks of the modulation of vibration, which peaks occur at twice slip frequency, it will be seen that each of these peaks corresponds to, and occurs at the same time as, a main peak of the modulation of stator current. Also each main peak of the modulation of vibration corresponds fairly closely to and occurs at about the same time as either a positive or a negative peak of the rotor current. This correspondence is not always exact. The phase differences may be due to the frequency modulation of the vibration as well as to the frequency modulation of the stator current.

2) The stator current is amplitude modulated at twice slip frequency (main peaks) and simultaneously at 6 times slip frequency (subsidiary peaks) and simultaneously at 12 times slip frequency (all peaks).

The amplitude modulation of stator current, like the modulation of vibration, is not absolutely regular and as explained above for the modulation of vibration this irregularity is believe to be a manifestation of frequency modulation of the stator current in the presence of the basic amplitude modulation of the stator current.

Also, as explained above in connection with the modulation of vibration, the modulation of stator current is such that the main peaks of the modulation of stator current occur at the same times as the main peaks of the modulation of vibration and at about the same times as the positive and negative peaks of the rotor current.

At the time the tests were conducted on Motor No---82 it was believed that the results were not in accordance with earlier observations. In particular the pulsations of the stator current ammeter needle appeared to be so fast that these pulsations could not be counted. Also the recorder charts were different from what was expected in that it was not immediately clear from these charts that the vibration and the stator current were each modulated at twice slip frequency. It was therefore decided to abandon the tests on Motor No.---82 and to conduct tests on Motor No.---79, hoping that the results of the tests on Motor No---79 would confirm the observations made on earlier dates or at least correspond more closely to them.

However, as the tests on Motor---79 proceeded and with the subsequent closer study of all the recorder charts for both motors it became clear that, although the conditions during the tests conducted on 22/3/77 may have been different to the conditions on earlier dates, the results obtained on 22/3/77 for both motors were basically the same as the results obtained by the simpler observations made with less sophisticated equipment at earlier dates. However the chart recordings made on 22/3/77 reveal much more detailed information than could be obtained from the simpler methods of observation at earlier dates. It was this wealth of information which in fact caused the initial confusion.

1800 KW 4 POLE 6600 V SLIPRING INDUCTION MOTOR DRIVING CENTRIFUGAL PUMP.

TESTS CONDUCTED AT LOWER PUMP STATION, UNDERGROUND, NO. 8 SHAFT ST. HELENA GOLD MINES LIMITED ON MOTOR NO---79 INSTALLED IN NO. 2 POSITION (MIDDLE POSITION).

DATE OF TESTS: 22/3/77.

NOTE These tests were done soon after the tests just described on Motor No---82.

INSTRUMENTS USED:

As described for tests conducted on Motor No---82 on 22/3/77.

EXPERIMENTAL DETERMINATION OF SLIP FREQUENCY:

The slip frequency was determined experimentally as described in the report of the tests conducted on 2/3/77.

The motor was stationary for some hours before this test was started. Before that it had been loaded. Therefore the motor was only slightly warm.

20 pulsations or cycles of rotor current were counted over a period of one minute. A second count also gave 20 pulsations of rotor current over a period of one minute.

The amplitude of oscillations of the rotor current were about the same as those observed earlier the same day on Motor No---82, but were rather weaker than had been observed in earlier tests. This is discussed under "Experimental Determination of Slip Frequency" for the tests conducted on Motor No---82 on 22/3/77.

STATOR CURRENT:

The stator current, as read on the existing ammeter installed on the main control panel in the pump chamber, was 138 amps. The ammeter needle was oscillating very slightly. It was possible to count 120 oscillations of the ammeter needle over a period of one minute. This corresponded to 6 x 20 cycles per minute or 6 times slip frequency. Refer "Experimental determination of Slip Frequency" above. At the time this observation was made it was believed that it conflicted with observations made on earlier dates. In the earlier tests the counting of the oscillations of an ammeter needle indicated that the stator current was amplitude modulated at twice slip frequency. However, the study which has been made of the recorder charts of the tests conducted on 22/3/77 has revealed the complexity of the stator current modulation and has shown that the results of the tests conducted on 22/3/77 are, in fact, consistent with the results of tests conducted on earlier dates because the stator current has been shown to be simultaneously amplitude modulated at twice, six times and twelve times slip frequency.

VIBRATION READINGS ON BEARING CAPS (STATOR CURRENT 138 AMPS):

NON-DRIVE-END BEARING:

ANGLE (DEGREES, REFER DIAGRAM)	RADIAL VIBRATION READING (MU)		PULSATION OF VIBRATION (PULSATIONS PER MINUTE)
	MINIMUM	MAXIMUM	
0	180	190	-
90	48	50	-
180	170	180	40
270	50	62	-

DRIVE-END BEARING:

ANGLE (DEGREES, REFER DIAGRAM)	RADIAL VIBRATION READING (MU)		PULSATION OF VIBRATION (PULSATIONS PER MINUTE)
	MINIMUM	MAXIMUM	
0	190	200	-
90	94	96	-
180	190	200	-
270	90	94	-

Polar diagrams of the above readings of vibration have been drawn but because the pulsations of vibration were counted only in one case polar diagrams for pulsation of vibration have not been drawn. The above vibration readings are higher than the readings of vibration on the bearing caps of this motor made on 8/3/77. For this reason the radial scale of the polar diagram has been chosen 1 mm = 4 MU as compared with scales 1 mm = 1 MU and 1 mm = 2 MU used before.

VIBRATION READINGS ON MOTOR FRAME (STATOR CURRENT : 138 AMPS):

POSITION	HORIZONTAL VIBRATION READING (MU)			
	TOP EDGE		AXIS	
	MINIMUM	MAXIMUM	MINIMUM	MAXIMUM
YA	-	-	175	180
YB	280	290	165	170
YC	-	-	160	165
YD	230	240	-	-
YE	-	-	125	130

Please refer to diagram showing positions where readings were taken and graphical plot of readings. The readings were taken in a similar way to the readings taken on the frame of Motor No---82 on 22/2/77 and 15/3/77 and the comments already made on those readings should be referred to.

This was the first occasion on which readings were taken of vibration on the frame of Motor No---79. The level of vibration was found to be generally much higher than the level of vibration on the frame of Motor No---82. The level of vibration on the top edge of the frame was higher than the level of vibration on the axis of the motor. The vibration on the bearings was read at a different time to the vibration on the frame so the bearing vibration readings have not been plotted on the graph. However a comparison with the readings of bearing vibration shows that the bearing vibration readings are about the same value or slightly higher than the vibration readings on the axis. This result contradicts the results obtained on Motor No---82 on 15/3/77, where the bearing vibration was found to be much lower than the vibration on the axis of the frame. This may be because the vibration on Motor No---79 is generally much higher than on Motor No---82 and possibly this higher level of vibration is an indication of something fundamentally different between the two motors in respect of the cause of vibration. On the other hand too much significance should not be given to readings which are not taken reasonably close together in time. A thorough investigation of frame vibration, if undertaken, could still show a common pattern. The vibration readings taken on the top edge of Motor No---79 are higher than the vibration readings on the bearings, as found before for Motor No---82.

COMMENTS ON RECORDER CHARTS:

The comments on the recorder charts applicable to Motor No---79 have already been made in the section dealing with the tests conducted on Motor No---82 on 22/3/77.

CONCLUSIONS:

As for Motor No.---82 already stated in the Section dealing with the tests conducted on Motor No---82 on 22/3/77.

1800 KW, 4 POLE, 6600 V SLIPRING INDUCTION MOTOR NO---80 FOR DRIVING CENTRIFUGAL PUMP, NO. 8 SHAFT, ST. HELENA GOLD MINES LIMITED. NO-LOAD CRITICAL SPEED TEST CONDUCTED ON THE TEST BED OF MESSRS. L.H. MARTHINUSEN LIMITED, DENVER, JOHANNESBURG.

DATE OF TESTS: 29/4/77.

BACKGROUND:

This motor had recently undergone repairs and overhaul at the Welkom, O.F.S. works of Messrs. L.H. Marthinusen Ltd., a firm specialising in motor repairs.

The motor had been taken out of service at the Lower Pump Station, No. 8 Shaft, St. Helena Gold Mines Limited because there had been a flashover on the sliprings of the motor and because the motor had been vibrating severely.

Once the motor had been dismantled at Messrs. L.H. Marthinusen, Welkom, it was found that the drive-end bearing inner ring had cracked in several places. One of these cracks had developed right across the inner bearing ring. This break in the ring had released the pressure of the ring on the shaft so destroying the interference fit of the bearing inner ring on the shaft. The slack ring had therefore worn the shâft down in the area of the ring's contact with the shaft. It was necessary to rebuild this worn area on the shaft and to refinish the surface of the shaft accurately in order to give the correct interference fit of the bearing inner ring on the shaft. Although the non-drive-end bearing and the shaft at the position of this bearing were not damaged, it was found, upon removing the non-drive-end bearing, that the shaft at the position of this bearing was under-sized to the extent that it did not give sufficient interference fit to this bearing. Therefore, it was necessary to rebuild this portion of the shaft as well as to refinish the surface of the shaft accurately at this position. The bearings on both the drive-end and on the non-drive-end were renewed. It may be noted this motor was not the first in this set of motors to suffer a cracked inner ring of the drive-end bearing.

It was also found that one of the rotor connections had cracked open. This was a clear crack and the cracked surfaces had sprung apart. The connection had not burnt open. Also, the rotor winding insulation resistance was found to have broken down to earth. The cracked connection was repaired and the rotor winding earth fault was cleared by repeated washing out and drying out by heating the rotor in an oven.

The development of fine cracks in the frame of the motor near the gussets at the motor feet were noted but these cracks were not repaired because repair by welding could distort the motor frame. Repair by metal stitching was considered but not proceeded with because it was believed that this would not be a solution to the problem of cracking. After such repairs of the cracks new cracks would develop through vibration of the motor in the future. The solution to the problem was believed to be in reducing the vibration rather than in patching up the cracks which were a symptom of the more basic vibration problem. It was believed that, pending a solution of the vibration problem, the cracks in themselves did not prevent safe and satisfactory operation of the motor.

Detailed speculation as to the causes and effects of the various failures which had occurred in this motor, as described above, will not be gone into here except to propose that it would appear possible that at least some of these failures were caused in part, if not wholly, by the initial excessive vibration which had developed in the motor, and that, upon their occurrence, at least some of these failures had contributed to a further increase in the vibration. It must be mentioned that in the case of the cracking of the inner ring of the drive-end bearing the possibility exists that the failure was caused or contributed to by the phenomenon known as "creep". This phenomenon is caused by insufficient interference fit between the bearing inner ring and the shaft. This poor interference fit permits a rolling and frictional action of the inner ring on the shaft which is known as "creep". This action takes place in the absence of lubrication between the surfaces concerned. Therefore excessive heat is developed, unequal thermal expansion occurs and this causes the bearing inner ring to crack. Poor interference fit would also reduce the bearing radial slackness and therefore contribute to dynamic eccentricity and so, through the action of unbalanced magnetic pull, to the development of the type of vibration which has been measured in this set of motors in earlier tests. The characteristics of this type of vibration has been discussed in detail in the reports of these earlier tests. To recall these characteristics briefly, this type of vibration is modulated at twice slip frequency and this type of vibration has a spectrum consisting mainly of a component at a rotational speed with a smaller component at twice rotational speed.

It was necessary to rebalance the rotor after the abovementioned repairs had been completed. Balancing facilities for such large motors do not exist in Welkom so it was necessary to send the motor to Messrs. L.H. Marthinusen's Denver Works near Johannesburg. It was decided to take this opportunity of conducting critical speed tests at Denver before the motor was rebalanced. This was facilitated by the fact that the test facility at Denver was capable of producing a variable frequency supply by means of a Schrage motor driving an alternator. The frequency range available was from about 21,8 Hertz to about 49,5 Hertz. It had been hoped to conduct critical speed tests into the overspeed region of the motor but no local testing installation was capable of providing the facility of operating the motor at overspeed, insofar as could be determined. This deficiency in local testing installations is surprising because BS 2613 : 1970 requires motors and generators to be capable of operation at an overspeed of 120 per cent of normal full and in certain cases (such as turbine driven equipment) to be capable of operation at even higher overspeeds. Certain motors imported from overseas have, in fact, been tested at the motor manufacturers works before dispatch at 120 per cent of normal full speed for two minutes. This has been done by simply running the motor (rated for a 50 Hertz supply) on a 60 Hertz supply.

INSTRUMENTS USED:

Vibration Meter:

This was a simple, purely mechanical, instrument of the clock gauge type for measuring vibration displacement. Made by THE L.S. STARRETT CO., ATHOL, MASS., U.S.A. Dial No. 196. Outer Body No. 192.
The circular scale was marked from 0 to 100 divisions, each division being 0,001 inch. This enabled readings of vibration displacement to be taken in thousandths of an inch, peak to peak, abbreviated to THOU in this report. This instrument was designed for measuring vertical vibration. It was therefore placed on top of the drive-end endshield and vibration in the vertical direction only was measured. A hasty attempt to mount the instrument so as to be able to measure horizontal vibration proved unsuccessful.

Tachometer:

A hand-held tachometer was used for measuring the rotational speed of the motor shaft.

Voltmeter:

A panel mounted indicating voltmeter was used for measuring the voltage applied to the motor terminals.

The excitation of the alternator was adjusted for different alternator speeds (frequencies) to give a constant voltage of 1600 volts applied to the motor terminals.

TEST CONDITIONS:

No-load, 1600 volts. The motor was standing free on the flat test bed rails, not bolted down.

CRITICAL SPEED TEST AT 1600 VOLTS:

MOTOR SPEED (RPM)	VERTICAL VIBRATION DRIVE-END ENDSHIELD PEAK TO PEAK (THOU)
642	0,1
672	0,2
696	0,1
720	0,3
742	0,2
760	0,2
775	0,5
802	2,0
822	1,5
847	2,0
867	2,0
888	5,0
902	4,5
918	2,5
940	1,5
958	1,0
988	1,0
1000	0,8
1018	0,8
1042	0,6
1055	0,5
1078	0,5
1100	0,6
1120	0,7

MOTOR SPEED (RPM)	VERTICAL VIBRATION DRIVE-END ENDSHIELD PEAK TO PEAK (THOU)
1140	1,0
1160	1,5
1170	1,8
1200	1,8
1212	1,8
1230	1,8
1245	1,9
1260	2,0
1280	2,0
1290	2,0
1310	2,2
1328	2,5
1342	2,8
1356	2,5
1368	2,5
1382	2,8
1395	3,0
1403	3,5
1412	3,5
1430	3,9
1446	4,0
1458	4,2
1464	4,5

CONCLUSIONS:

The test revealed a critical speed at about 888 r.p.m. It was not possible to run the motor at as high a speed as the rated full-load speed but for speeds just below rated full-load speed the vibration was climbing steeply so that at rated speed one could expect vibration of about the same value as was found at the critical speed of 888 r.p.m. It appeared, from the curve of vibration vs speed that was plotted, that one could expect a critical speed at about 2 x 888 r.p.m., that is, at about 1776 r.p.m. and that the vibration would be considerably higher at this critical speed than at about 888 r.p.m.

This situation emphasizes the importance of being able to test into the overspeed region. In this case an overspeed of 120 per cent of normal speed would have been 1,2 x 1485 R.P.M. = 1782 R.P.M. which would have been the approximate value of the higher critical speed.

The results of this critical speed test were considered so interesting that immediate arrangements were made to do further critical speed tests using the vibration analyser and chart recorder, and also to try to determine the stator frame resonant frequency by means of a storage oscilloscope.

1800 KW, 4 POLE, 6600 V SLIPRING INDUCTION MOTOR NO---80
FOR DRIVING CENTRIFUGAL PUMP, NO. 8 SHAFT
ST. HELENA GOLD MINES LIMITED.
NO-LOAD CRITICAL SPEED TESTS AND FRAME RESONANCE TESTS
CONDUCTED ON THE TEST BED OF MESSRS. L.H. MARTHINUSEN LIMITED
DENVER, JOHANNESBURG.

DATE OF TESTS: 2/5/77.

BACKGROUND:

Please refer to report on the tests conducted on this motor on 29/4/77.

INSTRUMENTS USED:

As for the tests conducted on 29/4/77, plus the following:-

Chart Recorder:

WATANABE MULTICORDER. The signal of vibration from the "DC Recorder" output of the vibration analyser was fed to this chart recorder.

Vibration Analyser:

IRD MECHANALYSIS MODEL 340 vibration analyser. The "DC Recorder" output of this instrument was fed to the chart recorder for the critical speed tests and the "oscilloscope" output of this instrument was fed to the storage oscilloscope for the motor frame resonance tests.

Storage Oscilloscope:

TEKTRONIX 464 storage oscilloscope. To measure resonance of the motor frame the signal from the oscilloscope output of the vibration analyser was fed to this storage oscilloscope.

TEST CONDITIONS FOR CRITICAL SPEED TESTS:

No-load. Tests were done for various voltages applied to the motor terminals. The motor was not bolted down but was standing free on the flat test bed rails. Some tests were done with wooden wedges driven below the foot pads where these projected beyond the test bed rails over the concrete floor of the test bed. Other tests were done with these wooden wedges removed so that the motor was standing completely free.

DISCUSSION OF CRITICAL SPEED TESTS:

These tests were conducted through the speed and voltage ranges available from the testequipment.

The measurements of vertical vibration taken with the Starrett vibration meter have been tabulated and plotted graphically. Where vibration values for a particular speed at different voltages have been equal to one another, the liberty has been taken, with the aim of making the graphs clear, to plot the vibration value corresponding to the higher voltage slightly higher. This is generally in line with the results measured by the Starrett vibration meter and is within the limits of experimental error.

These graphs show that with wedges used at the motor foot pads the critical speed is higher than when no wedges are used. The wedges add some additional support to the motor and the structure may be regarded as stiffer. A stiffer structure can be expected to have a higher critical speed. Also the use of wedges to stiffen the structure tends to reduce the vibration at all speeds. This effect is obscured only when the motor with wedges runs at a critical speed. In this case the value of vibration exceeds the vibration of the motor without wedge supports.

The results show that the effect of voltage is not as marked as the effect of the wedge support. As the voltage increases the level of vibration tends to increase at all speeds. As the voltage increases the critical speed is reduced and the peak vibration at critical speed increases. An increase in voltage causes an increase in the flux density which causes an increase in unbalanced magnetic pull. This increase in u.m.p. explains the effects which have been measured.

K.J. BINNS and M. DYE, in their paper "Identification of principal factors causing unbalanced magnetic pull in cage induction motors" PROC. IEE, Vol 120, No.3 MARCH 1973 state: "It is necessary to design a rotating machine so that the critical speed differs from the running speed by a safe margin, while, in some cases, keeping this margin as small as is reasonable for economic reasons. If u.m.p. is assumed to be proportional to eccentricity, its effect is equivalent to a negative component of shaft stiffness. Thus the critical speed of an induction motor is reduced by u.m.p. If the rate of change of pull with displacement is k_m, and the stiffness of the shaft is k_s, the reduction in critical speed is given by the factor $\sqrt{(1-k_m/k_s)}$, as has been shown by Rosenberg and Crawford. When a machine is operating at a speed near to a critical speed, vibrations will be present, and these give rise to a combination of both static and rotating eccentricity. Frohne is of the opinion that only disturbances in the airgap field caused by rotating eccentricity can affect the critical speed. .... A thorough investigation of the dynamic behaviour of a rotor operating near to a critical speed would seem to be needed before any of the methods of reducing u.m.p. can be disregarded as a means of minimising the effect on critical speed".

If Frohne's opinion quoted above is correct then the results of the critical speed tests forming the subject of this report would confirm the earlier tests done on this set of motors and the deductions therefrom, specifically, the main vibratory force occurs at rotational speed, the vibratory forces are modulated at twice slip frequency and vibratory forces at twice rotational speed occur but their magnitude is quite small, as discussed in greater detail in the writer's report on the tests which he conducted on 2/3/77. These phenomena all point to the occurrence of a dynamic eccentricity, that is a rotating eccentricity within the motor.

The recorder charts of vertical vibration vs motor speed have been annotated. A separate graph has been drawn effectively superimposing these charts. The results are similar to the information obtained from the Starrett vibration meter except that the effect of voltage, especially the records taken at 1600 volts and 2000 volts, are not in line with the trends indicated by the Starrett meter. Possibly this inconsistency may have been caused by some drift in the electronic equipment. In retrospect the vibration scale on the chart should have been chosen to give a bigger deflection.

In these tests the minimum critical speed occurred at 1110 r.p.m. with no wedges at the motor foot pads. On 2/5/77 the critical speed under the same nominal condition was measured at 888 r.p.m. This difference may be accounted for by a slight difference in the floor support for the motor on these two occasions. This illustrates the high sensitivity of this set of motors to the method of mounting and this confirms and reinforces previous experience on site.

As with the tests conducted on 2/5/77 there is every indication from the graphs of the results that a higher critical speed can be expected to occur at twice the first critical speed of 1100 to 1200 r.p.m., that is, at 2200 to 2400 r.p.m. (for these mountings). Also the resonance is not sharp (low Q factor). This can be seen from the first critical speed and also from the fact that the vibration has already begun to rise well below (about 200 r.p.m. below) the running speed, through a high value at the running speed to a very much higher value at the higher critical speed which is expected to occur at about 700 to 900 r.p.m. above the running speed.

Following these tests it was decided to conduct further critical speed tests with the motor bolted down.

CRITICAL SPEED TESTS:
MOTOR NOT BOLTED DOWN, WEDGES USED AT MOTOR FOOT PADS:

MOTOR SPEED (RPM)	VERTICAL VIBRATION ON DRIVE-END ENDSHIELD (THOUSANDTHS OF AN INCH PEAK TO PEAK) BY STARRETT VIBRATION METER FOR DIFFERENT VOLTAGES ON MOTOR TERMINALS			
	1600 VOLTS	2000 VOLTS	2500 VOLTS	3000 VOLTS
650	0,1			
670	0,1			
690	0,1			
710	0,1			
730	0,2			
750	0,2			
770	0,2			
790	0,3			
810	0,2			
830	0,3			
850	0,4			
870	0,5			
890	0,5			
910	0,6			
930	0,6			
950	0,7			
970	0,5			
990	0,7			
1010	0,7	1,0		
1030	0,8	1,0		
1050	0,8	1,2		
1070	0,9	1,0	1,0	
1090	0,9	1,0	1,2	
1110	1,0	1,2	2,0	
1130	1,4	1,5	3,0	
1150	1,6	2,0	5,5	
1170	3,5	4,0	4,0	
1190	3,5	4,0	4,0	
1210	3,5	3,5	3,5	
1230	3,0	3,0	3,5	
1250	2,5	2,5	3,0	
1270	2,0	2,5	2,8	
1290	2,0	2,5	3,0	3,0
1310	2,0	2,0	2,8	3,0
1330	2,0	2,0	3,0	3,0
1350	2,0	2,4	2,8	3,2
1370	2,0	2,4	3,0	3,2
1390	2,0	2,4	3,2	3,5
1410	2,0	2,5	4,0	3,4
1430	2,5	2,5	4,0	3,4
1450	2,6	3,0	4,0	4,0
1470	2,9	3,5	4,0	4,0

CRITICAL SPEED TESTS:
MOTOR NOT BOLTED DOWN, NO WEDGES USED AT MOTOR FOOT PADS:

MOTOR SPEED (RPM)	VERTICAL VIBRATION ON DRIVE-END ENDSHIELD (THOUSANDTHS OF AN INCH PEAK TO PEAK) BY STARRETT VIBRATION METER FOR DIFFERENT VOLTAGES ON MOTOR TERMINALS	
	1600 VOLTS	3300 VOLTS
650	0,1	
670	0,1	
690	0,1	
710	0,1	
730	0,3	
750	0,8	
770	0,8	
790	0,5	
810	0,5	
830	0,5	
850	0,5	
870	0,5	
890	0,5	
910	0,6	
930	0,6	
950	0,7	
970	0,7	
990	0,8	
1010	1,0	
1030	1,0	
1050	1,2	
1070	2,0	
1090	3,5	
1110	5,0	
1130	4,5	
1150	3,5	
1170	3,0	
1190	3,0	
1210	3,0	
1230	3,0	
1250	3,0	
1270	3,0	
1290	3,0	
1310	3,0	
1330	3,5	
1350	4,0	
1370	4,2	4,5
1390	4,2	4,6
1410	5,0	5,0
1430	5,5	5,5
1450	5,9	5,6
1470	6,0	6,0

These tests were conducted by hitting the motor main frame and endshields with a hammer. To determine the resulting resonant vibration the probe of the vibration analyser was was placed on the motor main frame or on one of the end-shields or on a bearing. The oscilloscope output of the vibration analyser was fed to the storage oscilloscope. The motor was not bolted down but was standing free on the flat test bed rails. There were no wedges below the motor foot pads.

With the probe on the top of the endshield to measure the vertical vibration a resonant frequency of about 500 to 550 Hz was measured. This component was ignored in later measurements and the predominant lower frequency components were concentrated upon and measured. Several measurements were made from which the following results emerged.

For vertical vibration measured at the top of the drive-end endshield resonant frequencies of approximately 40 Hz and approximately 50 Hz were measured. These vertical measurements were the first frame resonance measurements that were made. With experience it was possible in later measurements to show that a noise level at a frequency of 50 Hz existed. Possibly the vertical measurements recorded in this instance were merely the 50 Hz noise. On a future occasion if opportunity permits the remeasurement of the vertical resonance will be done.

With the probe placed horizontally on the main frame:

if the frame was hit more or less horizontally with the hammer a resonant frequency of approximately 20 Hz was measured.

if the DE or the NDE endshield was hit more or less horizontally with the hammer the resonant frequency measured was a mixture of approximately 20 Hz and approximately 40 Hz (double peaking effect).

With the probe placed horizontally on the NDE endshield or bearing:

if the NDE endshield was hit more or less horizontally with the hammer a resonant frequency of approximately 40 Hz was measured.

if the main frame was hit more or less horizontally with the hammer the resonant frequency measured was a mixture of approximately 20 Hz and approximately 40 Hz (double peaking effect).

With the probe placed horizontally on the DE endshield or bearing:

if the DE endshield was hit more or less horizontally with the hammer a resonant frequency of approximately 40 Hz was measured.

if the main frame was hit more or less horizontally with the hammer the resonant frequency measured was a mixture of approximately 20 Hz and approximately 40 Hz (double peaking effect).

1800 KW, 4 POLE, 6600 V SLIPRING INDUCTION MOTOR NO---80
FOR DRIVING CENTRIFUGAL PUMP, NO. 8 SHAFT.
ST. HELENA GOLD MINES LIMITED.
NO-LOAD CRITICAL SPEED TESTS AND FRAME RESONANCE TESTS
CONDUCTED ON THE TEST BED OF MESSRS. L.H. MARTHINUSEN LIMITED, DENVER, JOHANNESBURG.

DATE OF TESTS: 5/5/77.

BACKGROUND:

Please refer to report on the tests conducted on this motor on 2/5/77.

INSTRUMENTS USED:

As for the tests conducted on 2/5/77.

TEST CONDITIONS FOR CRITICAL SPEED TESTS:

No-load. Tests were done for various voltages applied to the motor terminals. The motor was bolted down.

DISCUSSION OF CRITICAL SPEED TESTS:

These tests were conducted through the speed and voltage ranges available from the test equipment.

Vertical Vibration:

The measurements of vertical vibration taken with the Starrett vibration meter have been tabulated and plotted graphically.

Recorder charts of vertical vibration vs motor speed were made during the tests. These have been annotated and a separate graph has been drawn effectively superimposing these charts.

All the results obtained show that by bolting the motor down the vertical vibration has been reduced considerably at all speeds.

There is no clear indication that a critical speed exists in the range of speeds through which the motor was tested.
There is some indication that a critical speed may be present just above 1300 r.p.m. The critical speed tests for horizontal vibration done later on 5/5/77 indicate a clear critical speed at 1310 r.p.m. Possibly the critical speed for vertical vibration, which seems to be suggested just above 1300 r.p.m. is associated with, or is induced by, the clear and strong critical speed for horizontal vibration at 1310 r.p.m. The effect of bolting the motor down has been to increase the first critical speed from 1100 or 1200 r.p.m. (as determined in the tests conducted on 2/5/77), with the motor not bolted down, to at least 1310 r.p.m., or possibly higher, with the motor bolted down.

It is possible that the increased stiffness resulting from bolting the motor down has increased the main critical speed to well above the normal running speed of the motor. The general upward trend of the curve below the running speed could be an effect of the broad resonance of the critical speed above the running speed.

The effect of voltage on the vibration with the motor bolted down is not as clear as in the readings of the Starrett vibration meter as was the case for the tests conducted on 2/5/77. The graphs drawn to superimpose the recorder charts show that increasing the voltage increases the vibration at higher speeds in the range of the speeds tested, but this relationship is confused in the region of about 1300 r.p.m., where a critical speed may exist. Further investigation would be necessary to clarify the effects of changes in voltage under the conditions of this test.

Horizontal Vibration:

The vibration analyser and chart recorder were used to determine the critical speed for horizontal vibration. The Starrett vibration meter was not used for these tests because this instrument was designed for measuring vertical vibration and could not be easily adapted to measuring horizontal vibration, as has been explained in the report of the tests conducted on 29/4/77.

Repeated tests, as recorded on the charts, indicated a clear critical speed at 1310 r.p.m. with the vibration very high at this critical speed, that is about 600 microns peak to peak as compared with only 10 to 20 microns peak to peak in the range of speeds from 650 to about 900 r.p.m. The tests also showed that the critical speed did not change for different voltages, from 1600 volts to 3050 volts, applied to the motor terminals.

The results of these horizontal vibration tests have revealed the following conditions that give cause for concern:-

1. That a critical speed should exist at all in the range of speed below the normal running speed of the motor. In larger 2 pole motors this condition is unavoidable but the structure of the motor is designed with the necessary stiffness and the critical speed is designed to be well away from the running speed of the motor. However, in a 4 pole motor of this size any critical speed should be well above the normal running speed.

2. That the critical speed is only 175 r.p.m. (or 11,8 per cent) below the rated full load speed of 1485 r.p.m.

3. That at the full load running speed the vibration has already begun an upward trend to a broad resonance major critical speed which probably exists at 2 x 1310 r.p.m. equal to 2620 r.p.m.

4. That at the full load running speed the vibration has a value of as much as one third of the value of the vibration at the first critical speed.

5. That these adverse conditions should exist in a motor properly bolted down onto test bed rails.

CRITICAL SPEED TESTS. MOTOR BOLTED DOWN:

MOTOR SPEED (RPM)	VERTICAL VIBRATION ON DRIVE-END ENDSHIELD (THOUSANDTHS OF AN INCH PEAK TO PEAK) BY STARRETT VIBRATION METER FOR DIFFERENT VOLTAGES ON MOTOR TERMINALS		
	1600 VOLTS	2800 VOLTS	3300 VOLTS
650	0,1		
670	0,2		
690	0,2		
710	0,2		
730	0,2		
750	0,2		
770	0,2		
790	0,2		
810	0,2		
830	0,2		
850	0,2		
870	0,2		
890	0,2		
910	0,2		
930	0,2		
950	0,3		
970	0,3		
990	0,3		
1010	0,3		
1030	0,3		
1050	0,3		
1070	0,3		
1090	0,3		
1110	0,4		
1130	0,4		
1150	0,4		
1170	0,4		
1190	0,5	0,5	
1210	0,5	0,5	
1220	-	0,5	
1230	0,7	0,5	
1240	-	0,6	
1250	0,8	0,6	
1260	-	0,6	
1270	0,9	0,7	
1280	-	0,8	
1290	1,0	1,2	
1300	-	1,1	
1310	1,5	1,3	
1320	-	1,6	
1330	1,0	1,7	
1340	-	1,7	1,2
1350	1,3	1,7	-
1360	-	1,6	1,3
1370	1,3	1,6	-
1380	-	1,5	1,3
1390	1,3	1,5	-
1400	-	1,5	1,4
1410	1,3	1,5	-
1420	-	1,5	1,4
1430	1,4	1,5	-
1440	-	1,4	1,4
1450	1,4	1,4	-
1460	-	1,4	1,4
1470	1,4	1,4	1,4

FRAME RESONANCE TESTS:

These tests were conducted in a similar manner to those conducted on 2/5/77, except that the motor was bolted down.

With the probe placed horizontally on the DE endshield:

When the DE endshield was hit more or less horizontally at various places with the hammer the following resonant frequency measurements were obtained:-

	22,85 Hz
	22,85 Hz
	24,38 Hz
	23,89 Hz
	24,17 Hz
	23,53 Hz
	23,53 Hz
AVERAGE:	23,6 Hz

With the probe placed horizontally on the main frame:

When the main frame was hit more or less horizontally at various places with the hammer the following resonant frequency measurements were obtained:-

	21,05 Hz
	22,86 Hz
	23,68 Hz
	22,86 Hz
AVERAGE:	22,6 Hz

With the probe placed horizontally on the NDE endshield:

When the NDE endshield was hit more or less horizontally at various places with the hammer the following resonant frequency measurements were obtained:-

	23,08 Hz
	25,45 Hz
	24,62 Hz
	23,08 Hz
	23,44 Hz
AVERAGE:	23,9 Hz

With the probe placed vertically on the DE endshield:

When the DE endshield was hit with the hammer a resonant frequency measurement of 46,67 Hz was obtained. It was determined by further measurements that 50 Hz noise was present. Further investigation of the vertical main frame and endshield resonances is necessary and will be done in the future if opportunity permits.

Further test with motor unbolted, standing free:

The above described horizontal main frame and endshield resonance tests showed that the endshield horizontal resonant frequencies were about 23 to 24 Hz. (motor bolted down), whereas, on 2/5/77, the endshield horizontal resonant frequency was found to be about 40 Hz (motor not bolted down).

It was therefore decided to make a check on the measurements of 2/5/77. The motor feet were unbolted and a horizontal resonant frequency of about 44 Hz was measured on the endshield. It was determined that this was definitely not 50 Hz noise.

CONCLUSIONS:

The accuracy of the above main frame and endshield resonance tests is certainly not better than $\pm$ 10 per cent. The measured average horizontal main frame resonance of 22,6 Hz (motor bolted down) corresponds to 1356 r.p.m. which is, within the limits of accuracy of the measurements the same as the measured critical speed of 1310 r.p.m. for horizontal vibration (motor bolted down). The measured horizontal resonances for the endshields (motor bolted down) though slightly higher than the measurements for the main frame are also, within the limits of accuracy of the measurements, the same as the measured critical speed for horizontal vibration (motor bolted down).

1800 KW, 4 POLE, 6600 V SLIPRING INDUCTION MOTOR DRIVING CENTRIFUGAL PUMP.

TESTS CONDUCTED AT LOWER PUMP STATION, UNDERGROUND, NO. 8 SHAFT, ST. HELENA GOLD MINES LIMITED ON MOTOR NO---79 INSTALLED IN NO. 2 POSITION (MIDDLE POSITION).

DATE OF TESTS: 17/5/77.

INSTRUMENT USED: IRD MECHANALYSIS PORTABLE VIBRATION METER MODEL NO. 306M SERIAL NO. BO. 8013641.

PLEASE NOTE ABBREVIATION: MU = μ = VIBRATION DISPLACEMENT READING, MICRONS PEAK TO PEAK.

BACKGROUND:

Please refer to the reports of the tests conducted on this motor on 8/3/77 and on 22/3/77. Also please refer to the reports of tests conducted on other motors in this set.

It was necessary to make measurements of vibration on 17/5/77 because it was intended to move this motor onto a new bedplate. This new bedplate had been made to the motor manufacturer's design and it was intended to have the motor manufacturer supervise the installation and grouting of the new bedplate. After installation of the motor on the new bedplate it was intended to take vibration measurements again for comparison. For this reason the aim ôf the tests conducted on 17/5/77 was to take as many vibration readings as possible on various parts of the motor in the time available. Therefore, on this occasion, only minor attention was given to the modulation of vibration and to the modulation of stator current, observations of rotor current were not made and no attempt was made to record these variables on charts as was done on 23/3/77.

TEST CONDITIONS:

The motor was coupled up to and driving the pump which was pumping with its delivery valve fully open. At the start of the tests the motor had already been running like this for 3 hours 20 minutes. Therefore the motor was well warmed up at the start of these tests.

STATOR CURRENT:

The stator current as read on the indicating ammeter on the motor control panel in the pump station was pulsating slightly but clearly from 145 to 148 amps approximately. This gives an average of 146,5 amps or 80,49 per cent of 182 amps which is the rated full load current of the motor. 45,5 pulsations of of the ammeter needle were counted over a period of one minute. A second count of the pulsations of the ammeter needle showed that there were 46 pulsations over a period of one minute. The pulsations of the ammeter needle were clearer and easier to count than was the case in earlier tests.

A reading of the stator current was taken again just before the vibration readings were taken on the non-drive-end bearing cap. This reading was 142/145 amps as compared to 145/148 amps read when the pulsations of stator current were counted. The differences between these readings given an idea of how the stator current was fluctuating slightly during the course of the tests. This fluctuation was a slow and irregular change and, as such, was different from the above described faster and regular pulsations of the stator current. The slow and irregular fluctuations of stator current may have been caused by changes in the supply voltage or by driven equipment load fluctuations or by conditions within the motor.

The voltmeter was situated in the substation, which was on a level above the pump chamber, and so only one reading of voltage was taken. It is therefore not possible to positively associate the fluctuation of stator current with changes in the voltage.

On simple theory the load imposed on the motor by the pump should have been constant but it is possible that some changes were taking place in the pumping system to cause changes in the pump shaft power requirement.

The fluctuations in the stator current could have been caused by a process originating from within the motor itself because the vibration readings taken on the motor were very high, having increased from lower values taken at earlier tests. This adverse condition of vibration may be symptomatic of some deterioration of the motor.

VOLTAGE:

The supply voltage measured in the substation was 6,5 kV.

PULSATIONS OF VIBRATION:

The pulsations of horizontal vibration were measured on the main frame of the motor half-way along the length of the main frame at the shaft height. 46 pulsations of vibration were counted over a period of one minute. This rate of pulsation was the same as for the rate of pulsation of stator current.

VIBRATION READINGS ON NON-DRIVE-END BEARING CAP (STATOR CURRENT 142/145 AMPS)

ANGLE (DEGREES, REFER DIAGRAM)	RADIAL VIBRATION READING (MU)	
	MINIMUM	MAXIMUM
0	185	193
11,25	190	195
22,5	165	170
33,75	170	175
45	150	160
56,25	85	90
67,5	58	60
78,75	38	40
90	32	34
101,25	80	88
112,5	90	98
123,75	138	145
135	170	178
146,25	180	190
157,5	205	215
168,75	215	225

ANGLE (DEGREES, REFER DIAGRAM)	RADIAL VIBRATION READING (MU)	
	MINIMUM	MAXIMUM
180	212	222
191,25	202	212
202,5	202	210
213,75	142	148
225	150	155
236,25	110	118
247,5	92	98
258,75	40	44
270	23	25
281,25	30	33
292,5	66	68
303,75	132	138
315	150	155
326,25	155	160
337,5	205	212
348,75	218	222

The polar diagram of the above readings has been plotted graphically. This polar diagram resembles two circular lobes, as found in the tests conducted on 8/3/77. A tracing has been made of the polar diagram superimposing two circles upon it. It can be seen that the graphical plots of the vibration readings lie approximately on these two circles, although the agreement is not as good as for the results of the tests conducted on 8/3/77. The properties of these two circles are the same as those given in the report of the tests conducted on 8/3/77 except that:-

1. the common tangent to the two circles is oriented at 78,5° and 258,75° (as compared with 112,5° and 292,5° as found on 8/3/77).

2. the diameters normal to the common tangent are oriented at 348,75° and 168,75° (as compared with 22,5° and 202,5° respectively, as found on 8/3/77).

The following questions arise:-

1. Why has the orientation of the axis of the lobes changed from that found on 8/3/77 to that found on 17/5/77?

2. What are the factors which cause the axis of the lobes to assume a particular orientation?

The answers to these questions are not known.

Because the orientation of the axis of the maximum vibration has changed it would suggest that the axis of the maximum vibratory forces has changed. Therefore, to the extent that these vibratory forces are associated with the u.m.p., the orientation of the axis of the maximum u.m.p. has changed. Why the orientation of the axis of the maximum u.m.p. should change is not known.

After the completion of the tabulated radial vibration readings on the non-drive-end bearing cap attempts were made to confirm or reproduce these readings. These attempts proved unsuccessful because the repeat readings were sometimes above and sometimes below the tabulated reading at a particular position (angle). Therefore, to find how the vibration was fluctuating over a period of time, vibration readings were taken, as tabulated below, at intervals over a period of 13 minutes for vertical vibration and over a period of 5 minutes for horizontal vibration. These readings have been plotted graphically. These readings are merely a sample of the fluctuation of the vibration over a short period of time. The writer's observations during this series of tests have led him to believe that much bigger fluctuations of vibration than those represented in these latest tabulations often occur. The causes of these fluctuations of vibration is not know but may be associated with the fluctuations of stator current as described and discussed earlier in this report. It is also noteworthy that the noise generated by the motor changes in two ways. The regular pulsations of vibration can often be associated with a regular change in the pitch and/or volume of the noise. The irregular fluctuation of vibration can also often be associated with a change in the sound of the motor. These sound effects have been determined by ear. Noise measurements have not been made in these tests but such noise measurements as a factor associated with the phenomena measured in the present series of tests could be the subject of a separate study.

The vibration readings measured during these tests on the non-drive-end bearing cap were higher than the measurements made on 22/3/77.

VIBRATION READINGS ON NON-DRIVE-END BEARING CAP:

TIME (HR-MIN)	VIBRATION READING (MU)			
	HORIZONTAL		VERTICAL	
	MINIMUM	MAXIMUM	MINIMUM	MAXIMUM
10 h 52			32	34
10 h 53			30	32
10 h 54			29	32
10 h 55			30	34
10 h 55,5			38	40
10 h 56			36	39
10 h 56,5			32	35
10 h 57			34	36
10 h 58			32	34
10 h 58,5			34	36
10 h 59			31	34
11 h 00			29	31
11 h 01			29	31
11 h 02			29	32
11 h 03			30	32
11 h 04			30	33
11 h 05			30	33
11 h 06				
11 h 07	220	225		
11 h 08	220	230		
11 h 09	210	225		
11 h 10	210	225		
11 h 11	210	220		
11 h 12	220	230		

VIBRATION READINGS ON DRIVE-END BEARING CAP:

ANGLE (DEGREES, REFER DIAGRAM)	RADIAL VIBRATION READING (MU)	
	MINIMUM	MAXIMUM
0	95	100
90	50	50

The vibration readings measured during these tests on the drive-end bearing cap were lower than the measurements made on 22/3/77.

It was not possible to take more readings than those recorded above because the stoppage of the pump was imminent and it was decided to take readings on the motor frame before the stoppage. These are the readings of horizontal vibration on the motor frame taken on the top edge of the motor frame at 11 h 15. The motor was stopped at 11 h 20.

The motor was restarted at 12 h 30 and readings were resumed at 14 h 10. These comprised readings on the non-drive-end and drive-end bearing caps and on the motor frame as tabulated and shown graphically on diagrams.

VIBRATION READINGS ON MOTOR FRAME:

POSITION	HORIZONTAL VIBRATION READINGS (MU)					
	TOP EDGE AT 11 H 15		TOP EDGE AT 14 H 30		AXIS AT 14 H 30	
	MINIMUM	MAXIMUM	MINIMUM	MAXIMUM	MINIMUM	MAXIMUM
ZA	-	-	-	-	100	105
ZB	-	-	135	140	100	110
ZC	-	-	130	140	100	105
ZD	175	185	135	145	-	-
ZE	170	180	145	155	-	-
ZF	195	210	170	180	120	130
ZG	230	240	180	190	-	-
ZH	250	265	200	210	140	150
ZI	-	-	-	-	190	200
ZJ	-	-	-	-	220	230

VIBRATION READINGS AT ABOUT 14 H 30 ON NON-DRIVE-END BEARING CAP:

ANGLE (DEGREES, REFER DIAGRAM)	RADIAL VIBRATION READING (MU)	
	MINIMUM	MAXIMUM
0	240	250
90	38	40
180	220	230
270	28	30

VIBRATION READINGS AT ABOUT 14 H 30 ON DRIVE-END BEARING CAP:

ANGLE (DEGREES, REFER DIAGRAM)	RADIAL VIBRATION READING (MU)	
	MINIMUM	MAXIMUM
0	100	105
90	58	62
180	100	110
270	30	32

VIBRATION READINGS ON MOTOR FRAME, BEDPLATE AND FOUNDATION:

POSITION	HORIZONTAL VIBRATION READING (MU)			
	DRIVE-END		NON-DRIVE-END	
	MINIMUM	MAXIMUM	MINIMUM	MAXIMUM
A	2	2,3	2,3	2,5
B	2	2	2,3	2,5
C	5	6	7	8
D	7	8	9	10
E	27	28	28	30
F1	46	48	62	64
F2	58	60	86	90
G	78	84	90	95
H	100	110	115	120
I	115	120	180	185
J	140	150	210	220
K	150	160	230	240
L	160	180	230	250

DISCUSSION ON VIBRATION MEASUREMENTS.

The vibration measurements taken on the bearing caps in the afternoon gave slightly increased values compared with the measurements taken in the morning.

The horizontal vibration measurements taken on the top edge of the frame of the motor in the afternoon were significantly lower than the measurements taken in the morning. This is in contradiction to the measurements made on the bearing caps. An explanation cannot be offered. Possibly some condition changed between the time the measurements were made on the bearing caps and the time the measurements were made on the frame.

Measurements were made on the frame of the same motor on 22/3/77 when it was found that the vibration on both the top edge and the axis of the frame reduced towards the non-drive-end of the motor. However the measurements made on 17/5/77 showed that the vibration on the top edge and on the axis of the frame increased towards the non-drive-end of the motor. Possibly this is due to a difference in mounting or alignment between these two dates, or a condition has changed within the motor.

Measurements made on the Motor No---82 on 15/3/77 revealed that the vibration at the bearing caps was much less than on the axis or the top edge of the frame of the motor. However, the measurements on Motor No.---79 on 17/5/77 showed that the drive-end bearing cap vibration was about the same as the nearby axis vibration, while the non-drive-end bearing cap vibration was considerably higher than the vibration anywhere else on the axis. Possibly different processes are taking place in the two motors. Whilst one may only be able to speculate as to the cause rather than being able to offer a sound explanation for these apparently contradictory phenomena, these peculiarities may be considered as evidence of the lack of stability of these motors.

The results of the horizontal vibration readings on the motor bedplate and foundation in the vertical plane through the motor holding down bolts are much the same as for the tests previously conducted and similar comments apply. Please refer to the discussion of the tests conducted on 8/3/77. A noteworthy difference from the results of earlier tests is the tendency of the graphs of the drive-end and the non-drive-end measurements to separate for readings progressing from the bottom to the top of the motor frame.

1800 KW, 4 POLE, 6600 VOLT SLIPRING INDUCTION MOTOR NO---80 FOR DRIVING CENTRIFUGAL PUMP, NO. 8 SHAFT ST. HELANA GOLD MINES LIMITED. NO-LOAD CRITICAL SPEED TESTS (INCLUDING OBSERVATIONS OF ROTOR CURRENT, MODULATION OF VIBRATION, MODULATION OF STATOR CURRENT) AND FRAME RESONANCE TESTS.

CONDUCTED ON THE TEST BED OF MESSRS. L.H. MARTHINUSEN LIMITED, DENVER, JOHANNESBURG.

DATE OF TESTS: 9/6/77.

BACKGROUND:

Please refer to the reports on the tests conducted on this motor on 29/4/77, 2/5/77 and 5/5/77. These tests were no-load critical speed tests in which the level of vibration was measured at different motor speeds. These tests also included frame resonance tests.

It was decided to combine such critical speed tests as had already been done on this motor with the types of measurement which had been made on 22/3/77 on Motors No---82 and No---79 installed underground. These measurements were chart recordings of the modulation of vibration, of modulation of stator current and of the rotor current taken while the motor was running on load.

The aim of the tests on 9/6/77 was to make observations of the behaviour of the modulation of vibration, of the modulation of stator current and of the rotor current in the vicinity of the critical speed with the motor running at no-load.

INSTRUMENTS USED:

As already reported for the tests conducted on 22/3/77, except for the following changes:-

Chart Recorder:

Rotor current (Red pen).
Modulation of vibration (Brown pen).
Modulation of Stator Current (Dark Blue pen).

Stator Current Instrumentation:

Instead of using the equipment installed underground a current transformer and ammeter on the test bed were used in a similar way to provide a signal through the Sample and Hold Circuit to the Chart Recorder to give a record of the modulation of stator current.

Sample and Hold Circuit:

The Sample and Hold Circuit used previously was suitable only for a fixed frequency supply (50 Hz). This Sample and Hold Circuit was modified by MR. T. CORFIELD, UNION CORPORATION GROUP ELECTRONICS LABORATORY to enable the modulation of stator current to be recorded for different supply frequencies ranging from about 20 Hz to about 50 Hz.

Rotor Current Instrumentation:

A shunt designed for d.c. use and rated 50 millivolts at 100 amps was used with a millivoltmeter. It was necessary to use a shunt of lower current rating than in the previous tests because the rotor current was lower with the motor on no-load than it was with the motor on load.

Tachometer:

The hand-held tachometer used in this and previous tests was a SUGAI KEIKI "TIME LIMIT" HAND TACHOMETER SER.NO.J177768, date 1968,made in Japan. Its dial was marked in revolutions per minute.

Voltmeter:

A panel mounted indicating voltmeter was used to measure the voltage applied to the motor terminals, as reported for tests conducted on 29/4/77. The excitation of the alternator was adjusted for different alternator speeds (frequencies) to give a constant voltage applied to the motor terminals for each test. The value of this applied voltage was changed from test to test as noted on the recorder charts.

Storage Oscilloscope:

TETRONIX 464 Storage Oscilloscope as used in the tests on 2/5/77 and 5/5/77 was used to measure the resonance of the motor frame.

TEST CONDITIONS:

The motor was bolted down for all the tests. The motor ran at no-load for the critical speed tests which were done for various voltages applied to the motor terminals as noted on the recorder charts. A complete key was inserted in the keyway in the motor shaft extension and the key was secured by fibreglass tape. This was in accordance with the manufacturers method of balancing the motor, that is, with a complete key in the keyway. Although not recorded in the reports of earlier tests a complete key was inserted in the keyway, as just described, in all of the earlier no-load critical speed tests.

The vibration analyser probe was secured to the drive-end endshield and horizontal vibration was processed through the vibration analyser, rectifier and filter circuit and chart recorder. The vibration analyser was set to the "Filter Out" position for all tests.

DISCUSSION OF TESTS:

These tests were conducted through the speed and voltage ranges available from the test equipment.

27 recorder charts were selected and annotated from a much greater length of chart recorded at the tests to give a reasonably concise but accurate record of what transpired at the tests.

The following are some general comments on these charts. For further details please refer to the charts themselves.

Chart No. 1:

Motor speed: 1465 r.p.m., Voltage on motor terminals : 3000 volts.

Modulation of stator current:

2 x slip frequency (main peaks).

Rotor Current:

Flat non-sinusoidal peak. This was not expected. No sound explanation can be given. A later test (Chart No. 21) at about the same conditions gave what seems to be a good sinusoidal wave. Possibly this change in wave-shape was caused by a change in the motor temperature.

Modulation of Vibration:

Very regular at 24 x slip frequency. This is different from results of earlier tests done on load where results of 2 x slip frequency (main peaks) or 8 x slip frequency (all peaks) and sometimes modulation at slip frequency was obtained.

Chart No. 2:

Motor speed : 650 r.p.m. Voltage on motor terminals : 1600 volts.

Modulation of stator current:

Irregular modulation but 7,3 Hz ripple, as for rotor current ripple, is present and this ripple intensifies near every zero and peak of rotor current.

Rotor Current:

This has a flat non-sinusoidal peak but a very heavy ripple of 7,3 Hz is present which ripple decreases and dies away with every main zero of the rotor current but increases again as the main rotor current wave increases. The cause of this ripple is not immediately clear. If the ripple were caused by the 5th space harmonic in the main wave flux the frequency induced by the 5th harmonic in the rotor would be

$$f_2 = f_1 \; (6-5S_1)$$

(ERNEST W. SUMMERS: "Vibration in 2-pole Induction Motors Related to Slip Frequency", AIEE Trans., Vol.74, Part III, 1954, pp. 69-72)

ignoring S_1 = slip of the rotor with respect to the fundamental because it is so small

and $f_1 = \dfrac{650 \text{ r.p.m.}}{1500 \text{ r.p.m.}} \times 50 \text{ Hz} = 21{,}667 \text{ Hz.}$

then $f_2 = 21{,}667 \times 6 = 129{,}99$ say 130 Hz and not 7,3 Hz.

A thorough study has not been done but it seems unlikely that other space harmonics can produce such a low ripple frequency.

Modulation of vibration:

This seems to be irregular.

Chart No. 3:

Voltage on motor terminals : 1600 volts. The motor speed was increased evenly but not accurately related to chart speed.

Modulation of stator current:

2 x slip frequency (main peaks).
12 x slip frequency (all peaks).

The frequency of modulation of stator current increases as the motor speed increases. The stator current has progressively decreased as the motor speed has increased.

Rotor current:

The slip frequency increases as the motor speed increases. Rotor current ripple appears and increases in value at about the critical speed.

Modulation of Vibration:

The value of vibration increases as the motor goes through a critical speed at approximately 1070 r.p.m. There is no clear pattern to the modulation of vibration.

Chart No. 4:

Voltage on motor terminals : 2000 volts.

As per Chart No. 3, except that there is even more ripple on the rotor current at about critical speed and at speeds just above critical speed. Simultaneous with the development of ripple on the rotor current is the appearance of ripple on the modulation of stator current. The rotor current has a distinct sharp non-sinusoidal peak.

Chart No. 5:

Voltage on motor terminals : 2500 volts.

As per Chart No. 4, but the rotor current wave-form is even sharper.

Chart No. 6:

Voltage on motor terminals : 3000 volts.

As for Chart No. 5, except that due to limitations of test equipment the lowest speed on the chart is already higher than the critical speed. Rotor current peaks are sharp but there is hardly any evidence of ripple.

Charts Nos. 7 to 15 inclusive:

This is a series of tests at 1600 volts in steps of motor speed to determine the critical speed accurately and to observe how the modulation of stator current, the rotor current, modulation of vibration and the level of vibration change for different speeds below, at and above the critical speed.

The modulation of stator current is found to be at 2 x slip frequency (main peaks) throughout these tests with irregular modulation at higher frequencies.

The rotor current has sharp non-sinusoidal peaks with varying degrees of sharpness throughout these tests. Ripple appears on the rotor current and this ripple becomes stronger at speeds just above the critical speed. The ripple is less strong at still higher speeds. At seemingly irregular times very intense ripple at higher frequency than the usual ripple appears on the rotor current and simultaneously on the modulation of stator current.

The modulation of vibration is regular at speeds lower than the critical speed but for each motor speed there is a particular relationship of modulation of vibration to the slip frequency. There is no fixed relationship of the modulation of vibration to the slip frequency which is true for all motor speeds, but the relationship applicable to one motor speed differs from the relationship at other motor speeds. As the motor speed increases and approached the critical speed the modulation of vibration loses its regularity.

The critical speed measured early in these tests as 1080 r.p.m. changed to 1050 r.p.m. later. Possibly this change in the value of the critical speed was caused by an increase in the temperature of the motor during these tests.

Charts Nos. 16 to 17:

This is a series of tests at 2000 volts similar to the tests just performed at 1600 volts (charts Nos. 7 to 15). The critical speed is found to be 1050 r.p.m. Evidence of ripple on the rotor current appears below the critical speed at 1010 r.p.m.

Chart No. 18:

This is a series of tests at 2500 volts. The critical speed is found to be 1040 to 1050 r.p.m.

Charts Nos. 19 & 20:

This is a series of tests at 2940 volts. The critical speed is found to be 1050 r.p.m.

Charts Nos. 21 to 23:

This is a series of tests at 1470 r.p.m. which is the highest speed attainable with this test equipment. The charts are taken at both 60 mm/min and 600 mm/min at 2940 volts, 2500 volts, 2000 volts and 1600 volts. No ripple is evident on rotor current. Modulation of stator current at 2 x slip frequency and higher frequency is present. Modulation of vibration is regular but seemingly unrelated to slip frequency. These charts are examples of the regular modulation of slip frequency at a motor speed which is somewhat higher than the critical speed.

Chart No. 24:

This is a test taken at 1600 volts 1180 r.p.m. with increased amplification of rotor current and at a chart speed of 600 mm/min. This is not much above the critical speed. The ripple on the rotor current can be clearly seen.

Chart No. 25:

This is a repetition of Chart No. 2. 1600 volts, 650 r.p.m., 600 mm/min. Rotor current ripple 7,2 Hz. (Chart No. 25) is about the same as 7,3 Hz (Chart No.2) but slip frequency has changed from 0,0113 Hz (Chart No.2) to 0,0077 Hz (Chart No. 25).

Charts Nos. 26 and 27:

Recordings at 600 mm/min with increased amplification of all 3 traces for 3000, 2500, 2000 and 1600 volts at 1050 r.p.m. which is critical speed or close to it. Ripple on rotor current is clearly evident at all voltages. Modulation of vibration is irregular.

FRAME RESONANCE TESTS:

The results of repeated tests are tabulated below.

PROBE		HAMMER BLOW		RESONANT FREQUENCY
POSITION	DIRECTION	POSITION	DIRECTION	
Top of DE end-shield.	Vertical	Top of DE end-shield.	Vertical	25 Hz for a few cycles, thereafter 30 Hz.
Top of Main Frame	Vertical	Top of Main Frame	Vertical	30 Hz.
Top of NDE end-shield.	Vertical	Top of NDE end-shield.	Vertical	18 Hz.
Side of DE end-shield.	Horizontal	Side of DE end-shield.	Horizontal	46 Hz.
Side of Main Frame	Horizontal	Side of Main Frame	Horizontal	17,5 to 20 Hz for a few cycles. Thereafter 45 to 47 Hz (difficult to resolve between this and 50 Hz).
Side of NDE end-shield	Horizontal	Side of NDE end-shield	Horizontal	45 to 47 Hz changing to 20 Hz and back to 45 to 47 Hz (double peaking).

CONCLUSIONS:

Rotor Current:

The behaviour of the rotor current is possibly one of the most interesting aspects of these tests. In the earlier tests done on loaded motors the wave-form of the rotor was always sinusoidal. In these present tests, however, the rotor current wave-form was not sinusoidal but displayed sharp peaks and almost triangular wave-shapes but flat peaks were also recorded. A ripple on the rotor current was evident at about critical speed and at speeds just above critical speed. This ripple is of a frequency which seems to be too low to be caused by the space harmonics in the main wave flux. At seemingly irregular times a very intense ripple at higher frequency than the usual ripple appears on the rotor current and simultaneously on the modulation of stator current. No explanation of these phenomena can be offered at this stage.

Modulation of Vibration:

These tests seem to indicate that the modulation of vibration is regular at speeds somewhat below and somewhat above the critical speed but the modulation of vibration becomes irregular at speeds in the region of the critical speed. There are some apparent exceptions to this so further tests throughout the speed range would be required to confirm this.

Even when the modulation of vibration is regular there is a lack of apparent correlation with slip frequency. This contrasts with a correlation found in the tests done on the loaded motors underground, that modulation of vibration was mainly at 2 x slip frequency or 1 x slip frequency.

Possibly this lack of correlation can be explained by the fact that this motor had recently been overhauled and that particular care had been taken to achieve a good interference fit between shaft and the inner ring of each bearing. This would have eliminated "creep" of the bearing inner ring on the shaft and would have reduced the bearing clearance and both these factors would have reduced the rotating eccentricity.

Modulation of Stator Current:

Modulation of stator current is consistently present at 2 x slip frequency (main peaks) in all test results. This is the same as was found in the tests on the loaded motors underground. This agreement of results is reassuring to a limited extent only, because this modulation of stator current now seems to be present while the modulation of vibration at 2 x slip frequency or 1 x slip frequency is not present. This seems to suggest that the modulation of stator current is related to slip frequency independent of what happens to the modulation of vibration.

Modulation of stator current at 12 x slip frequency (all peaks) is present at higher speeds from roughly critical speed up to full speed but this relationship is obscured by the appearance of other frequencies at speeds lower than about critical speed. At 1600 volts the modulation of stator current at 12 x slip frequency is present only at speeds somewhat higher than critical speed.

As already mentioned in connection with the rotor current, intense ripple of higher frequency than the usual ripple appears seemingly irregularly on the rotor current graph and simultaneously on the graph of the modulation of stator current.

General:

Possibly a deeper study of these graphs may reveal further conclusions. Clearly several avenues for further experimental work are suggested by these charts and by the conclusions already drawn.

1800 KW, 4 POLE, 6600 VOLT SLIPRING MOTOR NO---80 FOR DRIVING CENTRIFUGAL PUMP, NO. 8 SHAFT, ST. HELENA GOLD MINES LIMITED. VIBRATION TESTS CONDUCTED ON THE TEST BED OF MESSRS. L.H. MARTHINUSEN LIMITED, DENVER, JOHANNESBURG.

DATE OF TESTS: 28/6/77.

BACKGROUND:

Refer reports on the tests conducted on this motor on 29/4/77, 2/5/77, 5/5/77 and 9/6/77.

Since these tests the motor had been dismantled in order to remove the rotor for balancing. The rotor had been balanced in a balancing machine running at 375 r.p.m. The motor had then been re-assembled. The aim of the present tests was to check the vibration before returning the motor to the mine.

INSTRUMENTS USED:

Vibration analyser and oscilloscope as described in reports of the abovementioned earlier tests. In addition a resistance bridge instrument was used.

TEST CONDITIONS:

The motor was stood on the test bed floor but was not bolted down. A complete key was inserted into the keyway in the motor shaft extension and the key was secured by fibreglass tape. This was in accordance with the manufacturers method of balancing the motor, that is, with a complete key in the keyway. The motor was run at no load.

TEST RESULTS:

With the motor running at 3300 volts and 1470 r.p.m. the following horizontal vibration measurements were made:-

On drive-end bearing cap:	80 MU
On non-drive-end bearing cap:	130 MU

These results were clearly unsatisfactory. A frequency analysis of the vibration was done with the vibration analyser filter in the "broad" position, tuned to the frequency noted in the table below:-

FREQUENCY		HORIZONTAL VIBRATION (MU)	
C.P.M.	RELATION TO ROTATIONAL/CRITICAL SPEED	DRIVE-END BEARING CAP	NON-DRIVE-END BEARING CAP
2900	= 2 x Rotational Speed	25 Fairly Steady.	25 Fairly Steady.
1450	= 1 x Rotational Speed	60 Fairly Steady.	130 Fairly Steady.
790	= Critical Speed (Slightly more than half rotational speed).	25 Varying Randomly (Erratically)	20 Varying Randomly (Erratically)
TOTAL (FILTER OUT):		80	130

The test was repeated after the motor had been run for about an hour and it was therefore slightly warmer than it was for the last test. The following results were obtained:-

FREQUENCY		HORIZONTAL VIBRATION (MU)	
C.P.M.	RELATION TO ROTATIONAL/CRITICAL SPEED	DRIVE-END BEARING CAP	NON-DRIVE-END BEARING CAP
2900	= 2 x Rotational Speed	20 Fairly Steady.	28 Fairly Steady.
1450	= 1 x Rotational Speed	88 Fairly Steady	150 Fairly Steady.
830	= Critical Speed (Slightly more than half rotational speed).	20 to 25 Varying Randomly (Errationally)	20 to 25 Varying Randomly (Errationally)
TOTAL (FILTER OUT):		75	160

The total vibration was subject to a slow variation which may have been a pulsation at slip frequency or at 2 times slip frequency but this was not easy to determine because the slip frequency was so low.

The stator and rotor winding resistances were checked to determine if an electrical unbalance existed. A resistance bridge instrument was used. The following results were obtained at an ambient temperature of 14,5^{o}C:-

		Resistance (ohm)
Stator	R-Y	0,237
"	R-B	0,237
"	Y-B	0,237
Rotor	1-2	0,00978
"	1-3	0,00975
"	2-3	0,00975

There was no evidence of electrical unbalance in the above measurements.

The motor was run up to speed again and tripped to observe if the vibration was purely mechanical or if the vibration was excited by the electrical supply. The vibration analyser proble was applied to the non-drive-end bearing cap during this test. There appeared to be no change in vibration when the supply was tripped. This result appeared to contradict the results of earlier tests done on these motors installed in position underground when the vibration reduced sharply at the moment the motor tripped. Possibly this was because the motor was operating at no load in the latest test but was operating at partial load during the underground tests just before the supply was tripped.

In the present test the motor slowly retarted to rest after it was tripped. At roughly 805 to 810 r.p.m. the vibration increased and then reduced again indicating the presence of a critical speed. The response was rather flat in this region and judgement of the speed at which the peak occurred was difficult and it is possible that the critical speed may have been a bit higher than 810 r.p.m.

The above described trip and retardation test was repeated on the non-drive-emd bearing cap and the same results as before were obtained.

CONCLUSION:

It was decided to rebalance the motor in the fully assembled condition.

1800 KW, 4 POLE, 6600 V SLIPRING INDUCTION MOTOR DRIVING CENTRIFUGAL PUMP. TESTS CONDUCTED AT LOWER PUMP STATION, UNDERGROUND, NO. 8 SHAFT, ST. HELENA GOLD MINES LIMITED ON MOTOR NO---79 INSTALLED IN NO. 3 POSITION.

DATE OF TESTS: 18/8/77.

NOTE: In addition to reporting the tests conducted on 18/8/77 this report includes brief notes, where applicable, on tests conducted on 6/10/77. The tests conducted on 6/10/77 comprized a general survey of all the motors installed at both the Lower and Intermediate Pump Stations. It is not intended to submit any further report of the tests conducted on 6/10/77.

INSTRUMENT USED: IRD MECHANALYSIS PORTABLE VIBRATION METER MODEL NO. 306 M SERIAL NO. BO 8013641.

PLEASE NOTE ABBREVIATION: MU = μ = VIBRATION DISPLACEMENT READING, MICRONS PEAK TO PEAK.

BACKGROUND:

Refer reports of the tests conducted on this motor on 8/3/77, 22/3/77 and 17/5/77. Also refer to the reports of tests conducted on other motors in this set.

In the abovementioned earlier tests this motor was mounted on a bedplate designed, made and installed by the user. In the present tests the motor was mounted on a new bedplate designed by the motor manufacturer. The user had arranged for this new bedplate to be made. The new bedplate had been installed by the motor manufacturing staff, who had also lined up, installed and commissioned the motor on this bedplate. The present tests comprized vibration measurements on the motor mounted on the new bedplate for comparison with the vibration measurements taken in the tests conducted on 17/5/77. As in the tests conducted on 17/5/77 the aim at the present tests was to take as many readings as possible on various parts of the motor in the time available. Therefore, on this occasion only minor attention was paid to the modulation of vibration and to the modulation of stator current, no observations of rotor current were made and no attempt was made to record these variables on charts as was done on 22/3/77.

TEST CONDITIONS:

The motor was coupled up to the pump. The motor had only run about 4 hours since its installation in this position had been completed a few days before. The motor was stationary before the commencement of the present tests, having been used the previous day for 65 minutes. Therefore the motor was only mildly warm (about body temperature). The motor was then started and the tests were conducted with the motor driving the pump.

VIBRATION ZERO CHECK:

With the motor standing the following horizontal vibration readings were taken:-

On drive-end bearing:	0,2 to 1,2 MU
On non-drive-end bearing:	0,2 to 0,4 MU
On motor frame about 1/3 of motor length from non-drive-end bearing and at shaft height:	0,2 to 1,2 MU.

STATOR VOLTAGE:

The stator voltage as read on the voltmeter at the nearest substation on a level above the pump chamber was found to be varying between 6,65 and 6,75 kV. These readings were taken at about the time of the commencement of the tests. These readings were high compared with the rated voltage of 6,6 kV.

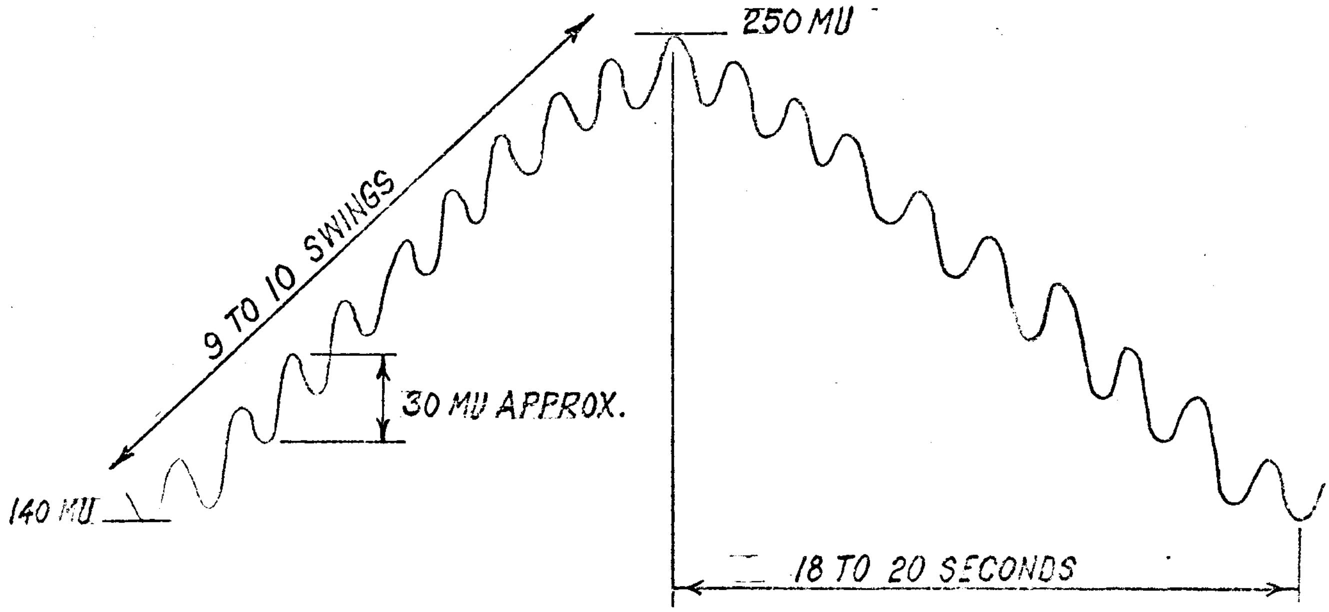
STATOR CURRENT:

The stator current was found to be fluctuating between 128 and 130 amps when read on the ammeter at the substation. This reading was taken at about the time of the commencement of the tests. A short while later the stator current was found to be fluctuating between 132 and 135 amps when read at the motor control panel.

PRELIMINARY READINGS OF VIBRATION AND OF VIBRATION MODULATION:

Just after the motor was started a "no load" reading was taken with the motor driving the pump but with the pump delivery valve still closed. This "no load" reading of vibration was taken on the non-drive-end bearing of the motor and the vibration was found to be 40 to 50 MU. As the pump delivery valve was opened and the load on the motor increased the vibration increased to approximately 250 MU and then fluctuated down to 150 to 200 MU.

The modulation of vibration was immediately seen to be different to the modulation observed in earlier tests. Modulation at presumably 1 or 2 times slip frequency (this could not be checked in the present tests) was superimposed on a slower cyclic variation in the general level of vibration. The total modulation was carefully checked and it was found that the vibration changed from a minimum of about 140 MU to a maximum of about 250 MU for 9 to 10 swings of the vibration meter needle. Each individual swing of the needle had a total (peak to valley) change of about 30 MU. The time from the complete cycle maximum to its minimum was measured twice and found to be 18 seconds and 20 seconds, as shown in the diagram below.



This pattern of modulation was found to be present on both bearings and on all positions of the motor frame at which measurements were made. The only change was one of scale, that is, the value of the vibration at different positions.

The writer made another visit to this pump station and took more readings of vibration on this motor on 6/10/77. It was found that the pattern of modulation was exactly the same as was found on 18/8/77, although the values of vibration had changed. Furthermore, a similar pattern of modulation was observed on 6/10/77 when measurements were made on Motor---78 installed in No. 1 Position at the Intermediate Pump Station. This suggests that this pattern of modulation is not necessarily associated with or attributable to the new bedplate and its method of installation. This does suggest that variations in the supply may be responsible or that a similar condition may be present in each motor. Unfortunately the time available for testing was limited on both these visits and recording equipment was not at hand. Further investigation of this type of modulation is required.

VIBRATION READINGS ON DRIVE-END BEARING CAP

ANGLE (DEGREES, REFER DIAGRAM)	RADIAL VIBRATION READING (MU)	
	MINIMUM	MAXIMUM
0	52	68
90	15	19
180	52	64
270	12	16

VIBRATION READINGS ON NON-DRIVE-END BEARING CAP

ANGLE (DEGREES, REFER DIAGRAM)	RADIAL VIBRATION READING (MU)	
	MINIMUM	MAXIMUM
0	90	170
90	34	36
180	100	200
270	14	19

DISCUSSION ON VIBRATION MEASUREMENTS ON BEARING CAPS:

The polar diagrams of the above readings have been drawn superimposed on the polar diagrams of the readings taken on 17/5/77.

It can be seen that the vibration measured on 18/8/77 is lower than that measured on 17/5/77. However, the vibration measured on 18/8/77 is higher than the limits specified on BS 2613 : 1970 and therefore the situation is unsatisfactory. Also, the motor was much cooler during the test done on 18/8/77 than it was during the test on 17/5/77. Earlier tests suggested that vibration increases with increase in temperature of the motor. The new pattern of modulation of vibration observed on 18/8/77 shows a much greater overall modulation than was present in the tests conducted on 17/5/77. It cannot be stated whether this reflects favourably or adversely on the new bed-plate and installation.

VIBRATION READINGS ON MOTOR FRAME:

POSITION	HORIZONTAL VIBRATION READING (MU)			
	TOP EDGE		AXIS	
	MININUM	MAXIMUM	MINIMUM	MAXIMUM
ZD	95	130	-	-
ZF	90	130	-	-
ZH	90	145	-	-

VIBRATION READINGS ON MOTOR FRAME, BEDPLATE AND FOUNDATION:

POSITION	HORIZONTAL VIBRATION READING (MU)			
	DRIVE-END		NON-DRIVE-END	
	MINIMUM	MAXIMUM	MINIMUM	MAXIMUM
A1	3	5	6	7
B1	8	16	11	17
C	5	9	7	12
D	6	10	8	13
E	17	24	16	29
F1	32	42	28	52
F2	33	46	32	58
G	56	78	45	74
H	63	89	68	110
I	75	135	78	125
J	90	160	85	138
K	110	163	90	143
L	85	125	90	143

DISCUSSION OF VIBRATION MEASUREMENTS ON MOTOR FRAME, BEDPLATE AND FOUNDATION:

These vibration measurements have been plotted graphically, superimposed on the graphs plotted for the measurements taken on 17/5/77. To avoid confusion measurements of vibration on the non-drive-end of the motor have been plotted for the higher positions on the motor only. In the lower positions on the motor the values for the non-drive-end are approximately the same as those on the drive-end.

The vibration on the foundation, bedplate and feet of the motor is higher than was measured on 17/5/77. The measurements of vibration at higher positions of the motor are mostly lower than was measured on 17/5/77. The modulation of vibration is much greater than was present on 17/5/77.

SHUTDOWN:

The motor ran from 10h00 to 11h35 when the pump had to be stopped because of the shortage of water. As was observed in earlier tests, the vibration fell away sharply to zero immediately the motor supply was tripped. The vibration meter needle then oscillated up and down 4 or 5 times before the vibration levelled to a very low value as the motor slowed down. As previously concluded this indicates that the vibration is electromagnetic in origin and not simply due to a mechanical unbalance condition.

4.3 TESTS ON 1800 HP (1340 KW), 4 POLE, 6600 V SLIPRING INDUCTION MOTORS:

1800 HP (1340 KW), 4 POLE, 6600 VOLT SLIPRING INDUCTION MOTOR NO---18/01. TESTS AT NO-LOAD CONDUCTED AT MESSRS. L.H. MARTHINUSEN LIMITED, DENVER, JOHANNESBURG, INCLUDING CRITICAL SPEED TESTS AT VARIOUS ECCENTRICITIES.

DATE OF TESTS: 28/6/77, 30/6/77 and 1/7/77.

BACKGROUND:

The oil lubricated, disc type, sleeve bearings of this motor had failed after about 100 running hours. When these bearings failed one of the shaft journals was severely damaged. The damage to the bearings and shaft was repaired and the motor was put back into service. After another 90 running hours the motor bearings and shaft failed again in almost the same way as before.

This motor became the subject of an investigation by the writer and others. This investigation included measurements of unbalanced magnetic pull. In order to make these measurements the motor endshields were modified enabling the rotor to be set to various values of stationary eccentricity. The writer took the opportunity of using this facility for adjusting the eccentricity to conduct the tests forming the subject of this report.

INSTRUMENTS USED:

Chart Recorder :

WATANABE Multirecorder.

The following signals were fed to this chart recorder:-

Rotor Current. (Red pen).
Modulation of Vibration. (Brown pen).
Modulation of Stator Current. (Black pen).

Vibration Analyser:

IRD MECHANALYSIS Model 340 vibration analyser. The d.c. output of this vibration analyser was fed to the Chart Recorder to give a record of the modulation of vibration.

This meant that the vibration signal was processed by the vibration analyser's internal rectifier and filter circuit. In previous tests done on other motors the oscilloscope output of the vibration analyser was fed through an external specially designed Rectifier and Filter Circuit to the Chart Recorder to give the modulation of vibration. However, as will be explained under "Stator Current Instrumentation" below, this external Rectifier and Filter Circuit was not available for this purpose because it became necessary to use this circuit to process the stator current signal.

Rectifier and Filter Circuit:

This circuit was designed and built by MR. T. CORFIELD, UNION CORPORATION GROUP ELECTRONICS LABORATORY. This circuit processed the signal of stator current. The output of this circuit was fed to the Chart Recorder to give a record of the modulation of stator current.

Stator Current Instrumentation:

A current transformer on the test bed was used to measure stator current. This was a Class AL, Ratio 100/50/20/10/5 amp C.T. The 50/5 amp ratio was used. The output of this current transformer was fed to a 5 amp ammeter on the instrument panel.

It was intended to feed the current transformer output to a Sample and Hold Circuit, which had been designed and built by MR. T. CORFIELD, UNION CORPORATION GROUP ELECTRONICS LABORATORY, in order to obtain the modulation of stator current. This Sample and Hold Circuit depended for its functioning on the zero crossing of the stator current. However, before the main tests were started, the stator current was observed on the oscilloscope and the presence of a ripple having a frequency of approximately 1800 Hz was observed to be superimposed on the 50 Hz fundamental frequency. It was considered that the true zero crossing of the 50 Hz fundamental would be obscured by this ripple and would lead to malfunction of the Sample and Hold Circuit. Therefore the use of the Sample and Hold Circuit was dispensed with and the Rectifier and Filter Circuit originally intended for the processing of the vibration was used in its place.

Therefore, the output of the current transformer (in fact, the voltage appearing across the ammeter terminals) was fed through the Rectifier and Filter Circuit to the Chart Recorder to give a record of the modulation of stator current.

Rotor Current Instrumentation:

A shunt of WESTON manufacture, designed for d.c. use and rated 100 amps 50 millivolts, was used with a 60-0-60 millivoltmeter connected across its terminals. A shunt of this low current rating was necessary to measure the rotor current under the no-load condition. The signal from the millivoltmeter terminals was fed to the Chart Recorder to give a record of the rotor current.

Tachometer:

A hand-held tachometer was used to measure the rotational speed of the motor shaft. This was a SAGAI KEIKI "TIME LIMIT" HAND TACHOMETER, SER. NO. J177768 DATE 1968, MADE IN JAPAN. Its dial was marked in revolutions per minute.

Voltmeter:

A panel mounted indicating voltmeter was used to measure the voltage applied to the motor terminals.

Oscilloscope:

A TEKTRONIX 464 Storage Oscilloscope was used to observe the stator current, rotor current and vibration.

SUPPLY:

The supply to the motor under test was provided by an alternator driven by a Schrage motor, these machines being part of the installed test bed equipment.

The excitation of the alternator was adjusted for different alternator speeds (alternator output frequencies) to give various constant values of voltage applied to the motor terminals. The value of the applied voltage was changed from test to test as noted on the recorder charts.

The fullest use was made of the range of voltages and frequencies which was available from the installed test bed equipment but this range of voltages and speeds was severely limited by the rating of the equipment as can be seen from an examination of the recorder charts.

TEST CONDITIONS:

The motor was stood on the test bed floor but was not bolted down. The tests done on 28/6/77 were done without wedges at the motor feet. Before the tests were started on 30/6/77 the motor feet were wedged with wooden wedges to improve the support at the motor feet.

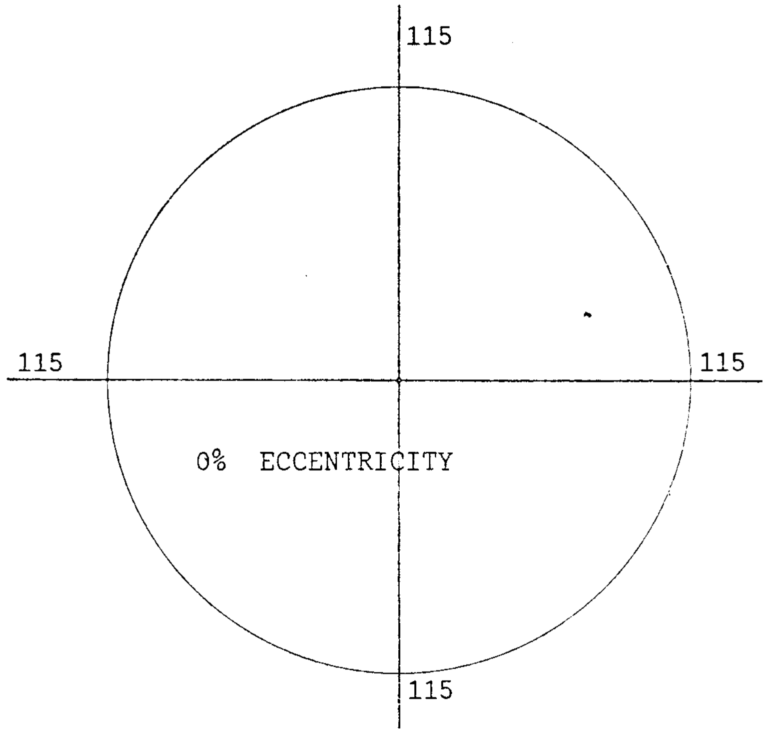
The motor was run at no-load throughout the tests.

DISCUSSION OF TESTS:

38 recorder charts were selected and annotated from the longer recordings made at the tests to give a reasonably concise but accurate record of what transpired at the tests. The following comments are intended to supplement the information provided on the charts. For further details please refer to the charts themselves.

Tests conducted on 28/6/77:

Before these tests were commenced the rotor position was adjusted and set to as accurate a state of concentricity with the bore of the stator as could be achieved. The airgap measurements were made at vertical top and bottom positions and at horizontal left and right hand posiions as shown in the diagram below. These measurements were made by feeler gauge and the airgap measurements are given in thousands of an inch (THOU). The rotor was adjusted until identical airgap measurements were achieved at each of the measurement positions at both the drive-end and the non-drive-end. It was found that it was possible to take the measurements at almost exactly the positions described despite the obstructing presence of the endshield arms at these actual positions. This method was simpler and at least as accurate as the method of taking the measurements at the 45° (diagonal) positions, which would have involved possible errors in the exact location of these 45° positions for each set of measurements as well as the complexity of calculating the airgap settings required at the 45° positions for each given eccentricity.



At the start of the tests the stator current was observed on the oscilloscope and the presence of a ripple of higher frequency than the fundamental supply frequency was found to be present, as has already been mentioned under "Stator Current Instrumentation" above. Using the oscilloscope in its storage mode the following measurements of the ripple frequency were made:-

1818 Hz, 1778 Hz, 1839 Hz, 1750 Hz.

The voltage on the motor terminals was 3300 volts. The motor speed was measured as 1465 r.p.m. The supply frequency was taken to be $\frac{1465 \text{ r.p.m.}}{1500 \text{ r.p.m.}} \times 50 \text{ Hz} = 48{,}83 \text{ Hz}$

Therefore the four measurements of ripple corresponded to 37,23, 36,41, 37,66 and 35,84 times the supply frequency respectively.

The three phase-to-phase output voltages of the alternator were examined on the oscilloscope with the alternator output open circuited. No ripple was found to be present in these supply voltages. These alternator output wave-forms appeared perfectly clean and sinusoidal and these oscilloscope displays were photographed.

The vibration tests were started with the probe of the vibration analyser vertical on the non-drive-end bearing. In addition to the modulation of vibration at 1 or 2 times slip frequency, a modulation of vibration at a much higher frequency of modulation was found to be present. This is shown on the recorder charts. A closer observation of the vibration on the oscilloscope showed that this modulation appeared to comprize two separate modulations with a phase difference between the two modulations.

A frequency analysis of the vibration was done using a stroboscope lamp triggered by the vibration analyser at its tuned frequency. The following results were obtained:-

F R E Q U E N C Y		VERTICAL VIBRATION			
		DRIVE-END BEARING CAP		NON-DRIVE-END BEARING CAP	
C.P.M.	RELATIONSHIP TO ROTATIONAL SPEED	DRIVE-END ANALYSER FILTER SETTING	MU	VIBRATION ANALSER FILTER SETTING	MU
2920	2 x ROTATIONAL SPEED	NARROW	8	BROAD	10
1460	1 x ROTATIONAL SPEED	BROAD	110	BROAD	62
TOTAL:		OUT	120	OUT	62

The above were the only discernable components of vibration. There was no component of vibration at about half the rotational speed or at about any critical speed as had been found in another motor. Also, the component at twice supply frequency, say about 97 to 100 Hz was especially sought but it was found to be negligible in magnitude, if present at all.

Charts Nos. 1 to 15 were recorded on 28/6/77. These tests were all done for vertical vibration on the non-drive-end endshield for a concentric condition of the airgap (zero eccentricity). No detailed comments will be made on these charts because the charts have been annotated and the peculiarities are clearly shown.

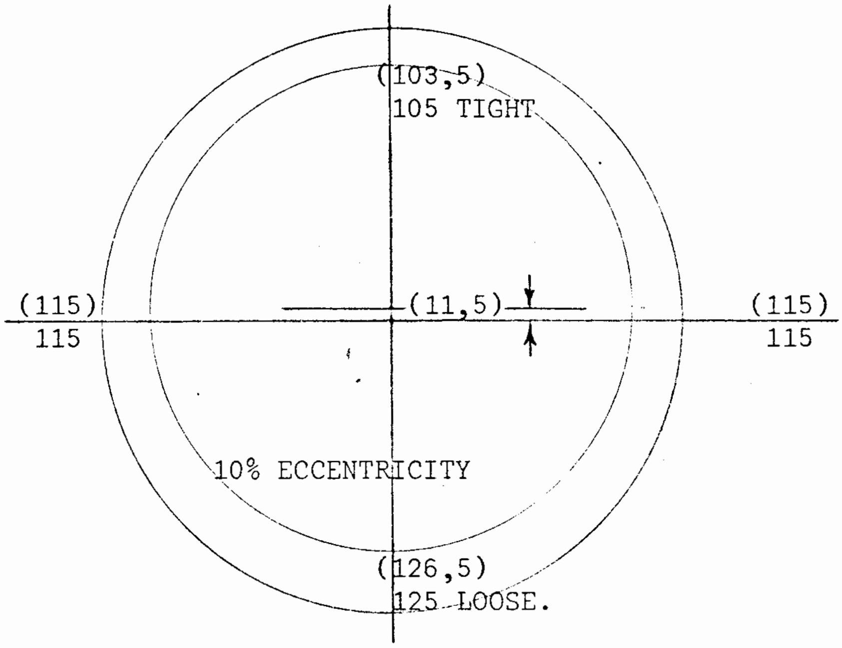
Tests conducted on 30/6/77:

The tests were resumed where they were left off on 28/6/77. The vertical vibration on the non-drive-end bearing was examined on the oscilloscope. It was found that the phase difference between the fundamental and the second harmonic components of vibration was constantly changing. The fundamental component of vibration had the frequency of rotational speed. The second harmonic component of vibration had the frequency of twice rotational speed.

The tests were then continued with the same basic conditions as appertained on 28/6/77 except that the probe of the vibration analyser was changed to the horizontal position on the non-drive-end bearing. Charts Nos. 16 to 21 were recorded.

The horizontal vibration on the non-drive-end bearing was examined on the oscilloscope and it appeared to be a fairly clean sine wave. This could be an anomoly because the horizontal vibration was high as compared to the vertical vibration and the horizontal vibration was distinctly modulated.

The airgap eccentricity was then changed from the concentric condition to a static airgap eccentricity of 10 per cent vertically upwards as shown in the diagram below:-



The figures in the diagram represent the dimensions at the positions shown and are expressed in THOU.

The figures in brackets are calculated ideal values.
The figures without brackets are measured values.
Charts Nos. 22 to 27 were recorded at 10 per cent eccentricity.

The ripple of about 7 Hz on the rotor current is shown on the recorder charts. It was noticed that there was a peculiar and distinct noise from the brushes on the slip-rings simultaneous with the appearance or increase of the rotor current ripple on the chart. This was at the lowest motor speed of about 750 r.p.m. at 1600 volts. It was found that this noise disappeared instantly when the supply to the motor was switched off.

It was noticed that solder had been thrown from the slipring connections. The rotor resistances were therefore checked giving the following results:-

Between sliprings	D-E : 0,03305 ohm
" "	D-F : 0,03305 ohm
" "	E-F : 0,03305 ohm
Across rotor terminals,	
including brushes	D-E : 0,04408 ohm
" " " "	D-F : 0,03938 ohm
" " " "	E-F : 0,04404 ohm

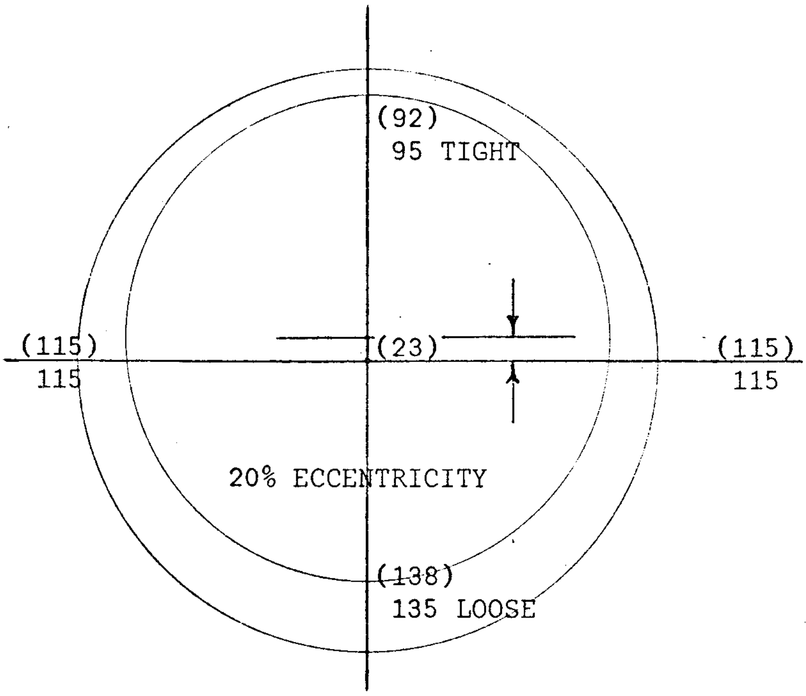
This showed the windings to be in order and balanced up to the sliprings but the brush-to-terminal resistances appeared to be unbalanced and possibly high.

The rotor currents were examined on the oscilloscope. Frequencies of about 7 Hz, a smaller component at about 125 Hz and a still smaller component at about 1000 Hz appeared to be present. The supply frequency was unfortunately not measured for this condition but the motor was running at about 750 r.p.m. (not measured).

A frequency analysis of vibration was made for the condition 10 per cent eccentricity, 1200 volts, 891 r.p.m. (critical speed). The fundamental frequency was the same as rotational speed. The second harmonic at twice rotational speed was present but could not be measured within the turning capability of the filter of the vibration analyser.

Tests conducted on 1/7/77:

The airgap eccentricity was changed from the 10 per cent static airgap eccentricity to 20 per cent static airgap eccentricity vertically upwards as shown in the diagram below:-



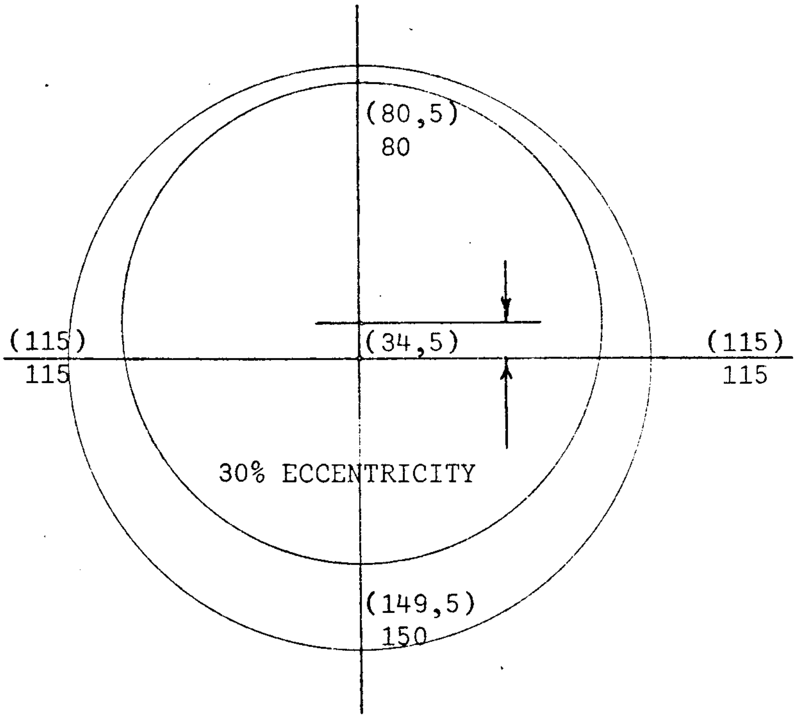
Charts Nos. 28 to 32 were recorded at 20 per cent eccentricity. It was noticed that the ripple on the rotor current was always 1/3 of the supply frequency. This was demonstrated on Charts Nos. 30 and 31 as follows:-

SUPPLY FREQUENCY (HZ)	ROTOR CURRENT RIPPLE FREQUENCY (HZ)
21,33	7,1
25,1	8,4
30,067	10,1

The rotor current was examined on the oscilloscope for the condition in which the motor supply frequency was 21,33 Hz (motor speed 640 r.p.m.) The rotor current ripple frequency was 7,1 Hz. The ratio "middle" frequency of rotor current to supply frequency was about 6 to 1.

On this basis the "middle" frequency of rotor current was about 6 x 21,33 Hz = 127,98 Hz. Also present was the "higher" frequency of rotor current. The ratio "higher" frequency of rotor current to "middle" frequency of rotor current was about 6 or 8 to 1. On this basis the "higher" frequency of rotor current was about 6 x 127,98 Hz = 768 Hz or 8 x 127,98 Hz = 1024 Hz.

The airgap eccentricity was changed to 30 per cent static air-gap eccentricity vertically upwards as shown in the diagram below:-



Charts Nos. 33 to 38 were recorded at 30 percent eccentricity. At this eccentricity the components of the rotor current were examined on the oscilloscope. At a motor speed of 753 r.p.m. at 1600 volts the supply frequency was 25,1 Hz and the rotor current ripple frequency was 84 Hz so the ratio supply frequency/ rotor current ripple frequency equal to 3 held in this case also. Furthermore the following relationships were demonstrated to hold very approximately:-

"Middle" frequency of rotor current = 6 x supply frequency
= 6 x 25,1 Hz = 150,6 Hz.
"Higher" frequency of rotor current = 6 x "middle" frequency
= 6 x 150,6 Hz = 90,3 Hz

The motor speed was increased to 902 r.p.m. at 1600 volts. The supply frequency was 30,7 Hz and the rotor current ripple frequency was 10,1 Hz so the ratio supply frequency/rotor current ripple frequency equal to 3 held in this case also. Furthermore the following relationships were demonstrated to hold very approximately:-

"Middle" frequency of rotor current = 6 x supply frequency
= 6 x 30,7 Hz = 184,2 Hz.
"Higher" frequency of rotor current = 6 x "middle" frequency
= 6 x 184,2 Hz = 1105,2 Hz.

The rotor current ripple appeared to predominate at the lower supply frequencies. This may be due to the response of the pen of the recorder. The ripple may well be present at higher supply frequencies in considerable strength even although this was not shown on the recorder charts. This theory could have been checked by oscilloscope examination of the rotor current at higher supply frequencies approaching 50 Hz but the possibility did not come to mind at the time of the test.

A notable feature of the tests at 30 per cent eccentricity was the instability of the vertical vibration for the 1600 volt condition. This occurred while attempting to find the critical speed. Refer Charts Nos. 33, 36, 37 and 38. This instability was not understood at the time of the test. It was thought that the airgap may have changed. The motor was stopped after the test and the airgap was checked at all the positions of airgap measurement and the previous measurements were found to be unchanged. It was also thought that the effect may have been caused by thermal expansion of the motor so changing the airgap even if by a small amount. However, the motor had been running at no-load. After the motor was stopped the temperature was checked at several positions by feel of hand and all positions were cool.

In order to obtain a clearer idea of the nature of the instability, the measurements recorded on Charts Nos. 36 and 38 have been tabulated separately and a graph has been drawn from this tabulation.

The following factors may be significant in any attempt to explain this instability:-

1. The general level of vibration was reduced considerably.

2. The change from the higher level to the lower level of vibration took place sharply during the course of the tests, as shown on the abovementioned graph.

3. The vibration vs. motor speed response curve appears to have become reduced in sharpness.

4. The value of the critical speed dropped considerably.

5. The sudden change in the vibration took place at the end of a long day of testing.

   The possible, although undetected, temperature rise could have caused thermal expansion.

6. The sudden change in vibration took place at the highest eccentricity tested.

   At higher eccentricity greater unbalanced magnetic pull can be expected resulting in bigger deflections of the shaft, rotor and stator structures. This would increase the probability of rubbing taking place between the rotating and stationary parts. The possibility of thermal expansion (refer 5 above) would increase the deflections and further increase the probability of rubbing. Rubbing need not necessarily take place only between the stator and the rotor laminations. It may be significant that the rubbing of the shaft on the non-drive-end bearing inner cover gave repeated trouble in other tests done on this motor.

7. The effect of vibration damping in reducing the level of vibration and in reducing the sharpness of the vibration vs. motor speed response curve is well known.

CONCLUSIONS:

In this series of tests the behaviour of the motor was studied under a number of different conditions.

The aspects of behaviour studied were:-

- 1. the vibration of the motor,
- 2. the modulation of this vibration,
- 3. the frequency analysis of this vibration,
- 4. the rotor current wave-form,
- 5. the stator current modulation and wave-form,
- 6. the value of critical speed.

The different conditions included:-

- 1. variations of eccentricity of the rotor in the stator,
- 2. variations of supply voltage,
- 3. variations of supply frequency (motor speed),

The main results of the investigation were:-

1. The vibration of the motor was measured on the non-drive-end bearing and was found to be high, more so in the horizontal than in the vertical direction.

2. Modulation of the vibration at twice slip frequency was found to be present with the rotor set concentrically (zero eccentricity) pointing to the existence of rotating (dynamic) eccentricities. Rotating eccentricity is known to cause a rotating magnetic attraction between the rotor and the stator causing a modulation of the vibration, as found. The modulation of vibration at twice slip frequency was also found to be present at 10 per cent eccentricity pointing to the existence of rotating eccentricities under this condition as well.

3. The frequency analysis of the vibration revealed that generally the main component of vibration was at the rotational speed of the motor and that the only other significant component was a smaller component at twice the rotational speed of the motor. This frequency analysis is indicative of rotating eccentricities associated with rotating magnetic attraction between the rotor and the stator. It was found that the phase difference between the fundamental and second harmonic component was constantly changing.

4. The rotor current wave-form was found to be severely distorted into various non-sinusoidal wave-forms. Also ripples were present on the rotor current wave-forms of a frequency of about 7 to 10 Hz which were shown to be almost exactly 1/3 of the supply frequency. These ripples were not of a frequency normally associated with harmonics of the space wave of magnetic flux due to mmf pattern set up by the stator windings, nor were the ripples of a frequency normally associated with the tooth ripples. Higher frequency components of the rotor current were also found.

5. The stator current was found to be modulated at twice slip frequency. Also a ripple of about 1800 Hz was found to be present on the stator current itself. When strong ripple appeared on the rotor current, ripple of the same frequency appeared on the modulation of stator current.

6. The value of critical speed was found to be reduced by increasing static eccentricity. There was no significant change in the value of the critical speed with change in voltage for the various eccentric conditions. For the concentric condition there appeared to be an unexpected tendency for the first and second critical speeds of horizontal vibration to increase with increasing voltage. The first critical speed of verticalvibration remained fairly constant with increasing voltage. Only one measurement of the second critical speed of vertical vibration was made. In retrospect it is regrettable that more measurements of critical speed were not made for all conditions in order to establish the trends more certainly, but the time available was limited.

The results of the series of tests has opened up several questions for further investigation.

EX CHART NO. 36.

MOTOR SPEED (RPM)	VERTICAL VIBRATION ON NDE BRG. (MU)
872	265
862	285
858	315
852	305
846	310
840	315
832	305
824	305
818	310
812	180
816	185
822	170
832	165
837	160
846	150
852	150
861	145
868	140
873	135
883	130
892	125

EX CHART NO. 38.

MOTOR SPEED (RPM)	VERTICAL VIBRATION ON NDE BRG. (MU)
807	180
821	170
831	160
841	155
851	150
862	140
847	150
832	165
818	180
802	195
793	215
772	235
762	255
752	260
742	270
732	270
717	260
702	255
690	215

CRITICAL SPEEDS (R.P.M.) EX TESTS DATED 28/6/77, 30/6/77 AND 1/7/77:

VOLTAGE (VOLTS)	CRITICAL SPEED (No)	ECCENTRICITY (%)			
		0	10	20	30
1200	1	-	H: 702 (27)	-	-
1200	2	-	H: 891 (27)	V: 881 (32) A: 877 (32)	V: 845 to 857 (34)
1300	1	-	H: 705 (27)	-	-
1300	2	-	H: 891 (27)	-	-
1400	1	-	H: 705 (27)	-	-
1400	2	-	H: 892 (27)	H: 870 (29)	V: 851 (33)
1500	1	-	H: 702 (26)	-	-
1500	2	-	H: 897 (26)	-	-
1600	1	V:1140 (8) H: 710 (18)	H: 750 (23)	-	-
1600	2	V:1440 (9) H: 925 (19)	H: 905 (24)	V: 881 (32) H: 878 (28) A: 862 (32)	V: 867 (33) 840 to 860 (36) 730 to 740 (38) (UNSTABLE)
1700	1	-	H:BELOW 760 (-)	-	-
1700	2	-	H: 897 (25)	-	-
1800	2	-	-	-	V:845 to 851 (34)
1890	2	-	-	-	V: 831 (35)
2000	1	V:1100 (10) H: 905 (21)	-	-	-
2000	2	H:1460 (11)	-	-	-
2500	1	H:1100 (13)	-	-	-
2500	2	H:1465 (12)	-	-	-
3000	1	NOT OBTAINABLE WITH THIS TEST EQUIPMENT.	-	-	-
3000	2	H:1465 (15)	-	-	-

H = HORIZONTAL VIBRATION; V = VERTICAL VIBRATION; A = AXIAL VIBRATION.
NUMBERS IN BRACKETS INDICATE RECORDER CHART No. APPLICABLE.

5. THEORETICAL ANALYSIS.

5.1 ANALYTICAL AIMS AND VIEWPOINTS:

In this thesis the writer is concerned with the radial forces present in induction motors and with the effects of these forces on motor reliability. The study of other forces, such as axial forces and torque-producing tangential forces falls outside the scope of this thesis. Also, for reasons which have been given elsewhere,[8] this thesis excludes the study of insulation problems and their effects on motor reliability.

Some forces will be applied from outside the motor. Other forces will be present within the motor as a result of the construction and operation of the motor. One or more of these forces may be applied to any component of the motor. Should the component concerned be too weak to withstand these forces, then the component will be damaged and will fail sooner or later, resulting in a minor or major failure of the complete motor.

Therefore, ideally, it is necessary to consider the motor as a whole, to assess all the forces that may be present and to calculate the ability of all the components to successfully withstand these forces over a long and useful motor life.

However, because of the complexity such a study would entail, simplifications have been made in the theory and in design procedure as will be shown later in this thesis. Because many serious premature failures of motors have occurred, particularly cases of motor bearing failures, cases of rotor rubbing on stator and cases of damage due to excessive vibration, a thorough re-examination of the theory and of the abovementioned simplifications would appear to be long overdue and urgently necessary. Such a critical review of the theory is the task the writer has set himself in this thesis.

The writer would classify the factors which should be embraced by a complete study of this field as follows:-

1. <u>Loading forces.</u>

    1.1. <u>Electromagnetic</u>

    including unbalanced magnetic pull and balanced magnetic forces, steady and vibratory.

    1.2 <u>Mechanical</u>

    including mechanical static loading and mechanical vibratory forces.

2. <u>Load carrying capacity.</u>

    2.1. <u>Mechanical</u>

    including bearing capacity

    2.2 <u>Structural</u>

    including strength of rotor and stator.

Whereas individual aspects of the above classification have been studied in the past, the writer visualises the development of a comprehensive and integrated theory of induction motor reliability.

The writer's aims and viewpoints are further stated as follows:-

1. To thoroughly investigate and define the basic assumptions made by the writer himself and by others in order to ascertain to what extent these assumptions are realistic and to enable later investigation to further develop the theory on realistic lines.

2. To investigate and develop the mathematical application of the harmonic field theory with reference to an induction motor having an eccentric airgap.

3. To provide a theoretical basis for the explanation of certain experimental observations.

4. To demonstrate mathematically the nature of the forces present in an induction motor having an eccentric airgap, including the unbalanced magnetic pull as well as balanced forces.

5. To investigate the influence of the forces on the design of the motor.

5.2 RADIAL LOADING FORCES IN AN INDUCTION MOTOR WITH AN ECCENTRIC AIRGAP:

In this section a study is made of the radial loading forces in an induction motor with an eccentric airgap. These may be called radial driving forces. These radial loading forces comprize two types of components:-

1. The radial electromagnetic forces resulting from the airgap flux distribution.

2. The radial mechanical forces resulting from mechanical unbalance. Static loading, which is important in motor design, in excluded from the study at this stage.

5.2.1 Radial electromagnetic force .

The attractive radial electromagnetic force per unit area between the magnetic surfaces forming an airgap in which the magnetic flux density is constant is [30]

$$F = \frac{B^2}{2\mu_0}\left(1 - \frac{1}{\mu_r}\right)$$

Assuming that the iron has infinite relative permeability[36] $(\mu_r = \infty)$,

$$F = \frac{B^2}{2\mu_0}$$

Where the radial magnetic flux density varies in position and in time, which is the case in the airgap of an induction motor,[33, p 68]

$$F_{\theta,t} = \frac{B^2_{\theta,t}}{2\mu_0}$$

Here the radial magnetic flux density $B_{\theta,t}$ and the attractive radial electromagnetic force $F_{\theta,t}$ depend on the periphery angle θ and on the time t.

5.2.2 Radial magnetic field.

is calculated from the relationship[43, p5][33, p8]

$$B_{\theta,t} = mmf_{\theta,t} \times \Lambda_{\theta,t}$$

where

$mmf_{\theta,t}$ represents the magnetomotive force (mmf). The mmf varies in position around the periphery of the bore of the motor and also varies in time. Therefore the mmf depends on the periphery angle θ and on the time t. The mmf may be expressed as a series of sine waves.

$\Lambda_{\theta,t}$ represents the airgap permeance. The airgap permeance varies in position around the periphery of the bore of the motor and also varies in time. Therefore the airgap permeance depends on the periphery angle θ and on the time t. The airgap permeance may be expressed as a constant (average) value and a series of sine waves.

The above relationship has been called the "Magnetic Ohm's Law".[49, p5]

The relationship is based on the assumption that all the mmf is expended in the airgap, that is, that the iron has infinite relative permeability $(\mu_r = \infty)$. It may be noted that this is the same simplifying assumption that was made in deriving the expression for the radial electromagnetic force.

In order to calculate $B_{\theta,t}$ the series of terms expressing $mmf_{\theta,t}$ is multiplied by the series of terms expressing $\Lambda_{\theta,t}$

Because each of the factors of $B_{\theta,t}$, that is, mmf$_{\theta,t}$ and $\Lambda_{\theta,t}$, is mathematically complex, the calculation of the product mmf$_{\theta,t}$ $\times \Lambda_{\theta,t}$ is extremely complex. Therefore, the previous investigators who have applied this expression have all made simplifying assumptions. The simplications that were made depended on the problem each investigator was faced with and upon his aim. The problems resulting from airgap eccentricity are only a class of the many problems arising from the complexity of the radial magnetic flux density. The Magnetic Ohm's Law has been the basis of attack on all these problems.

5.2.3 Harmonic radial magnetic fields.

The abovementioned complexity of the radial magnetic flux density has led to the representation of the radial magnetic flux density as an infinite series of harmonics of the flux density distribution defined as follows:- [26, p11]

> All harmonic radial magnetic fields are referred to the entire bore of the stator, that is, the "fundamental harmonic" of the order $\nu = 1$ forms one pole-pair. The periphery angle 2π corresponds to this pole-pair.
>
> If the motor has p pole-pairs, the harmonic radial magnetic field having p pole-pairs is called the "working harmonic". This harmonic is always of the order $\nu = p$. The periphery angle $\frac{2\pi}{p}$ corresponds to one pole-pair of this harmonic.
>
> An arbitrary harmonic of the ν th order forms ν pole-pairs. The periphery angle $\frac{2\pi}{\nu}$ corresponds to one pole-pair of this harmonic.

5.2.4 Main causes of harmonic radial magnetic fields: [49, p8]

In induction motors the main causes of the harmonic radial magnetic fields are as follows:-

1. Harmonic mmf's due to the winding distribution:-
   - 1.1 Phase-belt harmonics of stator and rotor.
   - 1.2 Slot harmonics of stator and rotor.
2. Variations in the airgap permeance caused by
   - 2.1 Slotting of the stator and rotor
   - 2.2 Eccentricity
   - 2.3 Saturation

5.2.5 Effects of harmonic radial magnetic fields: [49, p8]

1. Parasitic torques:-

   1.1 Asynchronous torques.

   1.2 Synchronous torques.

   1.3 Pulsating torques.

2. Unbalanced magnetic pull.

   2.1 Unbalanced magnetic pull depending on slot combinations.

   2.2 Unbalanced magnetic pull due to faults in the supply or in machine windings, called "unbalanced magnetic pull due to parasitic fluxes". Strictly, this is not an effect of the usual harmonic radial magnetic fields. Such unbalanced magnetic pull is caused by:-

      2.2.1 unsymmetrical supply to symmetrical machines.

      2.2.2 faults in the supply system (surge waves, short circuits).

      2.2.3 short circuited turns or coils and dis-symmetries in the magnetic circuit.

   2.3 Unbalanced magnetic pull caused by eccentricity and saturation which is independent of slot combinations.

3. Vibration.

4. Noise.

5. Voltage ripples.

5.2.6 Earlier calculations of the radial magnetic field and radial electromagnetic forces, a brief review:

Early investigators of the unbalanced magnetic pull (u.m.p.)(or "one-sided magnetic pull" as it has sometimes been called) produced in an induction motor with an eccentric airgap simply considered the resultant airgap flux density as it changed around the stator bore, the resultant airgap flux density being a maximum at the smallest airgap and a minimum at the largest airgap. The rotating harmonic magnetic fields were overlooked. This line of thought has been termed the [33, p6][13][34][5][3] "conventional approach". A true description of the forces present in an induction motor with an eccentric airgap was not possible on this basis. In particular, these early investigators concentrated on calculating "the" unbalanced magnetic pull. In essence, they assumed that this steady one-sided magnetic force was the adverse effect which had to be taken into account in the design of the motor.

The first demonstrations of harmonic fields were attributed [49, p 5] to STIEL (1919), FRITZE (1921) and CHAPMAN (1922). This line of thought has been termed the "rotating wave approach".[33 p 6] In view of the repeated references in the literature to harmonic fields and to harmonic field effects it may be more appropriate to designate this line of thought as "harmonic field theory". The application of this theory has produced outstanding results in the study of the parasitic effects accompanying the fundamental physical processes in induction machines.

Despite the limitations of the conventional approach and despite the distinguished record of the harmonic field theory over many years, the influence of the conventional approach has nevertheless been most persistent. The conventional approach concept of unbalanced magnetic pull is still the basis of design practice. The persistent influence of the conventional approach is evident even in the theoretical field. As an example, when the harmonic field theory was introduced by FREISE and JORDAN (1962)[23] into the study of the electromagnetic relationship in the eccentric airgap of an induction motor it was with a view to calculating the unbalanced magnetic pull alone. Other forces, steady and varying, were neglected. A knowledge of the magnitude of the unbalanced magnetic pull due to the eccentricity of the rotor was considered to be an essential requirement in the design of the machine construction because the unbalanced magnetic pull was regarded as having a considerable effect on the critical speeds of the machine, and therefore the study was directed to this end. As recently as 1977, HELLER and HAMATA[26, 7, p 71-76], in their book, submitted an outline of the work of FREISE and JORDAN, amplified in respect of the effect on saturation by the work[5][6] of BRADFORD (1968), BRADFORD also being influenced by the conventional approach. Even this recent work of HELLER and HAMATA is clearly directed to the calculation of the unbalanced magnetic pull alone, neglecting other forces.

RAI's thesis[33] in 1973 presented a diligent study of the effects of airgap eccentricity. Influenced by the conventional approach he calculated formulae for the unbalanced magnetic pull with multiplying factors to allow for the effect of slotting of the stator and rotor cores, the saturation of the iron and the presence of damper windings or other parallel paths in the stator or rotor windings. However, he was very concious of the effects of flux harmonics. He experimentally observed the vibratory forces and what he believed to be, modulations of these forces. He calculated the variation in magnitude of the vibratory force with time in the case of static eccentricity. Considering the vibratory forces present in the case of dynamic airgap eccentricity he regarded the rotational component at full speed to be similar to the u.m.p. occurring with static eccentricity, that it resulted from a similar mechanism and could be calculated in the same way as the u.m.p. for static eccentricity. He provided physical explanations for experimentally observed phenomena, particularly the frequencies ofvibration and the modulations of vibration but he did not go on to prove or develop these ideas mathematically.

Amongst the most notable applications of harmonic field theory has been the investigation of unbalanced magnetic pull depending on slot combinations. A great amount of outstanding work has been done in this field leading to the development of rules for slot combinations which should be avoided. Simplifying assumptions were made by the investigators into these effects. Using the Magnetic Ohm's Law to calculate the radial magnetic field these investigators used any of the following approaches:-

1. mmf harmonics x constant permeance. [49, p8]

2. the fundamental mmf wave x permeance harmonics. [49, p8]

3. selected mmf harmonics x selected permeance harmonics. [2, pp325f]

The study of unbalanced magnetic pull due to eccentricity has generally been considered separately from the study of u.m.p. depending on slot combinations.

The subject of saturation has been studied with regard to its effect on the u.m.p. due to eccentricity on the one hand and with regard to its effect on the u.m.p. depending on slot combinations on the other hand. It is considered that saturation produces further harmonic fields. [49, pp 14-15]

5.2.7 Mmf wave:

It will be assumed, for the purpose of the present analysis, that if the stator has a symmetrical series - connected three phase winding the stator currents produce an mmf wave given by [33, p 52]

$$\mathrm{mmf}_{\theta,t} = M_1 \cos(p\theta - \omega t - \phi)$$

The angular symbols in this expression are defined in Fig.1.

Other authors have made different assumptions.

FREISE and JORDAN (1962) [23] effectively take

$$\mathrm{mmf}_{\theta,t} = -M_1 \cos(p\theta - \omega t - \phi)$$

This merely involves a change of sign, which consequently appears in the expression for $B_{\theta,t}$ calculated from this expression for $\mathrm{mmf}_{\theta,t}$. This work is repeated in HELLER and HAMATA's book (1977). [26, pp 71-76]

MEILER, SPERLING and TIKVICKI (1973) [31] take

$$\mathrm{mmf}_{\theta,t} = M_1^I \cos(p\theta - \omega t - \phi) + M_2^I \cos(p\theta + \omega t - \phi)$$

The first term, $M_1^I \cos(p\theta - \omega t - \phi)$, is a foreward rotating component. The second term, $M_2^I \cos(p\theta + \omega t - \phi)$ is a relatively small backward rotating component. The authors consider it very important to take this backward rotating component into account because they believe that it can, in conjunction with static eccentricity, lead to vibration oscillation of twice supply frequency. On this basis they produce, for a 2 - pole machine, an expression for a type of u.m.p. derived from a component having a frequency of double supply frequency for static eccentricity. This is in addition to the usual steady component of unbalanced magnetic pull. This may be true, but the writer will not adopt this refinement in the present study because:-

1. The writer's approach is to develop the theory on the basis of simpler assumptions. On this basis he proposes to investigate the theory more deeply, more rigidly and further than appears to have been done before. The development of the theory on the basis of more complex assumptions would be the subject of a more advanced study.

2. The abovementioned additional component of unbalanced magnetic pull is small, being only $\frac{M_2^I}{M_1^I}$ of the usual component of unbalanced magnetic pull. M_2^I is stated by the abovementioned authors to be small relative to M_1^I

3. The writers analysis later in this thesis, on the basis of the above stated simpler assumption,

$$mmf_{\theta,t} = M_1 \cos(p\theta - \omega t - \phi)$$

will demonstrate the existence of a component of vibration of twice supply frequency which is $\frac{1}{\epsilon}$ times the magnitude of the usual component of unbalanced magnetic pull, therefore greater than the usual component of unbalanced magnetic pull because $\epsilon < 1$ and usually $\epsilon \ll 1$

In favour of the abovementioned authors outlook is the fact that any component of unbalanced magnetic pull will have a deforming effect on the stator and rotor different from that of a balanced vibration.

It is, however, noteworthy that, despite the authors' approach via the harmonic field theory, they are influenced by the conventional approach to direct their efforts to the calculation of the unbalanced magnetic pull alone and they do not go on to investigate the other forces present.

At this stage it is interesting to consider the normal expression for mmf harmonics:- [2,p91]

$$\begin{aligned} mmf_{\theta,t} &= M_1^{II} \cos(\theta - \omega t - \phi) + M_5^{II} \cos(5\theta + \omega t - \phi) \\ &+ M_7 \cos(7\theta - \omega t - \phi) + M_{11}^{II} \cos(11\theta + \omega t - \phi) \\ &+ M_{13} \cos(13\theta - \omega t - \phi) \end{aligned}$$

This expression has been used by investigators into unbalanced magnetic pull caused by slot combinations. [2,p330]

However, this expression is a simplication based on a uniform airgap, that is, a perfectly concentric airgap. The mmf's of all poles and of all phases are assumed to act on flux paths of the same permeance. Therefore, all the even harmonic terms and triple harmonic terms cancel out from the more general expression for the mmf. In the case of a motor with an eccentric airgap the mmf's of all poles and of all phases no longer act on flux paths of the same permeance. Therefore, the even harmonic terms and triple harmonic terms no longer cancel out completely but appear in the general expression with reduced magnitude. Thus we have:- [2,p88]

$$\begin{aligned} mmf_{\theta,t} &= M_1^{III} \cos(\theta - \omega t - \phi) + M_2^{III} \cos(2\theta - \omega t - \phi) \\ &+ M_3^{III} \cos(3\theta - \omega t - \phi) + \text{----} \quad \left[\text{FOREWARD REVOLVING MMF'S}\right] \\ &+ M_1^{IV} \cos(\theta + \omega t - \phi) + M_2^{IV} \cos(2\theta + \omega t - \phi) \\ &+ M_3^{IV} \cos(3\theta + \omega t - \phi) + \text{----} \quad \left[\text{BACKWARD REVOLVING MMF'S}\right] \end{aligned}$$

This is, therefore, the general expression for the mmf applicable to a motor with an eccentric airgap. The application of this expression would form the subject of a more advanced study.

5.2.8 Airgap variations:

In practice the airgap length in a motor is not uniform as idealised in elementary theory. There are notable variations from the mean value of the airgap length due to the following factors:-

1. Slotting of the stator and the rotor.

2. Airgap non-uniformity due to:-

   2.1 constructional inaccuracies, for example, stator bore not round.

   2.2. deformations of the stator and/or rotor, for example, nodular deformation of the stator and/or rotor due to electromagnetic forces.

   There is no eccentric displacement between the centre of the rotor and the centre of the stator with this type of airgap non-uniformity.

3. Airgap non-uniformity caused by eccentric displacement between the centre of the rotor and the centre of the stator. In this case it is assumed in analysis that the surfaces of the stator and the rotor are each perfectly cylindrical. This is usually called "airgap eccentricity" and can be due to:-

   3.1 mechanical causes, for example, inaccurate positioning of the rotor with respect to the stator.

   3.2 electromagnetic causes, for example, eccentricity caused by unbalanced magnetic pull depending on slot combinations.

The above factors, 1, 2 and 3 may occur each alone or one or more together in different combinations.

The literature does not distinguish between the factors 2 and 3 above and includes 2 in 3 even although this is strictly not [49,p12] correct. This is evident, for example, in VON KAEHNE's useful list of the origins of "eccentricities" as he calls them. Furthermore it appears that the factor 2 has not been analysed theoretically. Factor 1 has been [49,pp8-11] deeply studied by past investigators. Factor 3 has been analysed [23][33,p51][26,p72] theoretically and this analysis will be developed in this thesis. [49,pp11-13][30]

5.2.9 Airgap eccentricity and slot harmonics:

The present study concentrates on eccentricity of the airgap. Particularly this ignores permeance harmonics arising from slot combinations. There are three reasons for adopting this procedure:-

1. This procedure follows the work of previous investigators of the effects of airgap eccentricity.

2. The motors on which experimental observations were made by the writer did not have undesirable slot combinations. The undesirable slot combinations are those slot combinations which should be avoided to prevent adverse effects such as vibrations according to the slot rules derived by investigators of unbalanced magnetic pull depending on slot combinations.

3. This procedure may be regarded as the inverse of the procedure of investigators of unbalanced magnetic pull depending on slot combinations who ignored the effects of airgap eccentricity.

2.10 Stationary and rotating eccentricities:

Airgap eccentricity comprizes two different types of eccentricity which may occur alone or together:-

1. Stationary eccentricity: [49, p12]

This is sometimes called "static eccentricity". [33, p29] Stationary eccentricity occurs if the centre of the rotor is the centre of rotation. A stationary eccentricity is stationary in space. In Fig. 1 the rotor rotates about 0 . 0′, δ_{min} and δ_{max} are stationary in space.

2. Rotating eccentricity: [49, p12]

This is sometimes called "dynamic eccentricity". [33, p31] Rotating eccentricity occurs if the centre of the rotor is not the centre of rotation. The centre of the stator is the centre of rotation. The centre of the rotor rotates around the centre of the stator. In Fig. 1 the rotor does not rotate about 0′, but 0′ itself rotates about 0 at the angular velocity ω_ϵ.
δ_{min} and δ_{max} progress around the periphery of the stator bore at a speed $\frac{D}{2}\omega_\epsilon$, where D is the diameter of the stator bore.

Taking ω_R = angular velocity of the rotor. (This is the nominal angular velocity of the rotor if it were rotating about its own centre).

[49, p12]
We have $\omega_\epsilon = K\,\omega_R$

For K = 0 $\omega_\epsilon = 0$ This gives stationary eccentricity.

For K = 1 $\omega_\epsilon = \omega_R$ The eccentricity rotates at the angular velocity of rotation of the rotor.

For $K > 1$ $\quad \omega_\epsilon > \omega_R$ The eccentricity rotates with an angular velocity faster than that of the rotor. This case occurs with eccentricities caused by unbalanced magnetic pull due to two harmonic fields which can be avoided by slot rules. This case will not be considered in this thesis because the motors on which experimental observations were made by the writer did not have undersirable slot combinations.

2.11 Size of eccentric airgap:

FREISE and JORDAN, [23] RAI, [33,p51] HELLER and HAMATA [26,p72] consider the size of the eccentric airgap to be given by

$$\delta_{\theta,t} = R - r - e\cos\alpha$$

where, e = eccentric displacement of the centre of the rotor from the centre of the stator.

and $\alpha = \theta - \omega_\epsilon t - \phi_\epsilon$ as defined in Fig. 1.

As can be seen from the above relationship the size of the airgap $\delta_{\theta,t}$ depends on the periphery angle θ and the time t.

This expression for $\delta_{\theta,t}$ is an approximation.

The exact value of $\delta_{\theta,t}$ is calculated in Fig. 2:-

$$\delta_{\theta,t} = R - \sqrt{r^2 - e^2\sin^2\alpha} - e\cos\alpha$$

From MACLAURIN's series it can be shown that

$$\sqrt{r^2 - e^2\sin^2\alpha} = r\left[1 - \frac{1}{2}\frac{e^2}{r^2}\sin^2\alpha - \frac{1}{8}\frac{e^4}{r^4}\sin^4\alpha - \text{----}\right]$$

Neglecting the term $\frac{1}{8}\frac{e^4}{r^4}\sin^4\alpha$ and terms of higher order,

$$\delta_{\theta,t} = R - r\left[1 - \frac{1}{2}\frac{e^2}{r^2}\sin^2\alpha\right] - e\cos\alpha$$

$$= R - r - e\cos\alpha + \frac{1}{2}\frac{e^2}{r}\sin^2\alpha$$

This shows that the airgap is larger than the value assumed by the above-mentioned authors' by the amount $\frac{1}{2}\frac{e^2}{r}\sin^2\alpha$

Taking this further, $\frac{1}{2}\frac{e^2}{r}\sin^2\alpha = \frac{1}{2}\frac{e^2}{r}\left[\frac{1}{2} - \frac{1}{2}\cos 2\alpha\right]$

$$= \frac{1}{4}\frac{e^2}{r} - \frac{1}{4}\frac{e^2}{r}\cos 2\alpha$$

The application of this expression would form the subject of a more advanced study. The writer will use the expression $\delta_{\theta,t} = R - r - e\cos\alpha$ in his analysis.

If δ is the mean airgap length then R-r = δ

Also, from Fig. 1, $\delta + e = \delta_{max}$

$$\delta - e = \delta_{min}$$

Therefore, $\delta = \dfrac{\delta_{max} + \delta_{min}}{2}$ [31]

Substituting, $\delta_{\theta,t} = \delta - e\cos\alpha$ [33,p5]

Therefore, $\delta_{\theta,t} = \delta(1 - \epsilon\cos\alpha)$ [33,p5]

or $\delta_{\theta,t} = \delta\{1 - \epsilon\cos(\theta - \omega_\epsilon t - \phi_\epsilon)\}$

where, the relative eccentricity $\epsilon = \frac{e}{\delta}$

For stationary eccentricity, $\omega_\epsilon = 0$ and $\delta_{\theta,t} = \delta\{1 - e\cos(\theta - \phi_\epsilon)\}$

According to the relationship $\delta_{\theta,t} = \delta\{1 - \epsilon\cos(\theta - \omega_\epsilon t - \phi_\epsilon)\}$;

for rotating eccentricity the airgap variation with time will be the same in all radial directions, that is, for all periphery angles θ. There would be a phase difference in the time variation of the airgap length from one radial position to another. For an mmf wave of constant magnitude this can be expected to produce forces of the same magnitude in all radial directions. In this case there would be a phase difference in the time variation of the forces from one radial position to another. For a motor constructed equally stiff in all radial directions the vibration produced would be the same in all radial directions, although there would be a time phase difference in the vibration from one radial position to another.

This condition is different to the writer's experimental observations. It was found that, during normal operation of the motors tested, the vibration was much greater in the horizontal direction than in the vertical direction. When carefully measured the line of maximum vibration was inclined at a slight angle to the horizontal. The greater magnitude of the horizontal vibration could possibly be explained by greater frame stiffness in the vertical than in the horizontal direction. However, there is a possibility that the variation in airgap length may have been substantially different to that postulated in the above expressions for $\delta_{\theta,t}$. For example, with significant bearing clearance and with the shaft resting on the bottom of the bearing more horizontal than vertical movement of the rotor may have been taking place.

It is therefore interesting to note that LANDY (1975) [30] considers a case in which the vibration in the horizontal direction was much greater than in the vertical direction. His approach to the calculation of the airgap length is different to that of other investigators. As a first approximation he considers displacement of the rotor axis to take place in the horizontal direction only and hence deduces that the length of the airgap at any position θ can be expressed as

$$\delta_{\theta,t} = \delta\left(1 - \epsilon \cos \omega_f t \cos\theta\right),$$

assuming that the horizontal displacement between the rotor and the stator axis varies at some angular velocity ω_f.

This expression for $\delta_{\theta,t}$ states, in effect, that there exists a time dependent modulation of the position dependent airgap length.

From this expression, when $\cos \omega_f t = 0$ then $\delta_{\theta,t} = \delta$. This condition would take place when the centre of the rotor coincides with the centre of the stator during the horizontal oscillation of the rotor.

On the other hand, by definition of the conditions leading to the expression $\delta_{\theta,t} = \delta\left\{1 - \epsilon \cos(\theta - \omega_\epsilon - \phi_\epsilon)\right\}$, the centre of the rotor can never coincide with the centre of the stator unless e = o, so this condition cannot arise with constant rotating eccentricity.

However, these two expressions can be combined, as follows:-

$$\delta_{\theta,t} = \delta\left\{1 - \epsilon \cos(\theta - \omega_\epsilon t - \phi_\epsilon)\right\}$$

for rotating eccentricity (FREISE and JORDAN) [23]

$$\delta_{\theta,t} = \delta\left(1 - \epsilon \cos \omega_f t \cos\theta\right)$$

for horizontally oscillating eccentricity (LANDY) [30]

Now let $\delta_{\theta,t} = \delta\left[1 - \epsilon_0 \cos(\theta - \omega_\epsilon t - \phi_\epsilon) - \epsilon_1 \cos \omega_f t \cos\theta\right]$

The eccentricity comprizes two components,

ϵ_0 representing stationary (non-oscillating) eccentricity and rotating eccentricity.

and $\epsilon_1 \cos \omega_f t$,representing horizontally oscillating but non-rotating eccentricity.

The total eccentricity ϵ would be a vectorial addition of these two components as shown in Fig. 3.

In the expression $\delta_{\theta,t} = \delta\left[1 - \epsilon_0 \cos(\theta - \omega_\epsilon t - \phi_\epsilon) - \epsilon_1 \cos\omega_f t \cos\theta\right]$

for $\omega_\epsilon t + \phi_\epsilon = 0$ (that is, stationary eccentricity ϵ_0 in the horizontal direction)

we have $\delta_{\theta,t} = \delta\left[1 - \epsilon_0 \cos\theta - \epsilon_1 \cos\omega_f t \cos\theta\right]$

$$= \delta\left[1 - (\epsilon_0 + \epsilon_1 \cos\omega_f t)\cos\theta\right]$$

and $\epsilon = \epsilon_0 + \epsilon_1 \cos\omega_f t$

Also, in the expression $\delta_{\theta,t} = \delta\left[1 - \epsilon_0 \cos(\theta - \omega_\epsilon t - \phi_\epsilon) - \epsilon_1 \cos\omega_f t \cos\theta\right]$

when $\epsilon_0 = 0$ (no stationary or rotating eccentricity).

we have $\delta_{\theta,t} = \delta(1 - \epsilon_1 \cos\omega_f t \cos\theta)$

This is the same as LANDY's [30] expression for horizontally oscillating eccentricity.

and when $\epsilon_1 = 0$ (no horizontally oscillating eccentricity)

we have $\delta_{\theta,t} = \delta\left\{1 - \epsilon_0 \cos(\theta - \omega_\epsilon t - \phi_\epsilon)\right\}$

This is the same as FREISE and JORDAN's [23] expression for rotating eccentricity.

If in $\delta_{\theta,t}$ we put $\omega_f = \omega_\epsilon$

then $\delta_{\theta,t} = \delta\left[1 - \epsilon_0 \cos(\theta - \omega_\epsilon t - \phi_\epsilon) - \epsilon_1 \cos\omega_\epsilon t \cos\theta\right]$

These expressions would provide a more accurate representation of the size of airgap in many horizontal spindle induction motors than the expressions in earlier use. The application of these expressions would be the subject of a more advanced study.

5.2.12 Permeance wave:

From the expression, airgap length, $\delta_{\theta,t} = \delta - e\cos\alpha$, FREISE and JORDAN (1962) [23] derived an expression for the permeance wave, as follows:-

The permeance wave $\Lambda_{\theta,t} = \dfrac{1}{\delta_{\theta,t}} = \dfrac{1}{\delta - e\cos\alpha} = \dfrac{1}{\delta(1-\epsilon\cos\alpha)}$

Putting $\frac{1}{\delta} = \Lambda_0$, they obtained $\Lambda_{\theta,t} = \Lambda_0\,(u + v\epsilon\cos\alpha)$

The meanings of u and v will be explained presently.

The writer would state these relationships more correctly, as follows:-

$$\Lambda_{\theta,t} = \frac{\mu_0 a}{\delta_{\theta,t}} \quad \text{webers per ampere turn [37, p 26]}$$

where a = area of airgap in square meters

μ_0 = permeability of free space

$= 4\pi \times 10^{-7}$ Henrys per metre

Therefore, $\Lambda_{\theta,t} = \dfrac{\mu_0 a}{\delta - e\cos\alpha} = \dfrac{\mu_0 a}{\delta(1-\epsilon\cos\alpha)}$

Putting $\dfrac{\mu_0 a}{\delta} = \Lambda_0$, $\Lambda_{\theta,t} = \Lambda_0\,(u + v\epsilon\cos\alpha)$

In the writer's work $\Lambda_{\theta,t}$ and Λ_0 are defined in this way. RAI (1973) repeats the expressions of FREISE and JORDAN. [33, p9] [23] u and v were each defined by FREISE and JORDAN as a function of the eccentricity and the values of u and v were given in a graph showing their dependence on ϵ. For $\epsilon = 0$, $u = v = 1$. Both u and v increased more than linearly as ϵ increased. RAI (1973) reproduced this graph in his thesis. [33, FIG 12]

FROHNE (1967 to 1969) [33, p 10] represented the airgap permeance by a Fourier series as

$$\Lambda_{\theta,t} = \Lambda_0\left\{\Lambda_{01} + \Lambda_1\cos\alpha + \Lambda_2\cos(2\alpha)\;---\right\}$$

where

$$\Lambda_{01} = \frac{1}{\sqrt{1-\epsilon^2}}$$

and
$$\Lambda_n = \frac{2\left(1-\sqrt{1-\epsilon^2}\right)^n}{\epsilon^n\sqrt{1-\epsilon^2}}$$

This gives
$$\Lambda_1 = \frac{2\left(1-\sqrt{1-\epsilon^2}\right)}{\epsilon\sqrt{1-\epsilon^2}}$$

HELLER and HAMATA (1977) [26,p72] repeat this with some changes in symbol nomenclature. Taking these changes of nomenclature into account they state that in the range $\epsilon < 0.7$ the following approximations hold:-

$$\Lambda_{01} \doteq \frac{1}{1-\frac{\epsilon^2}{2}}$$

$$\Lambda_1 \doteq \frac{\epsilon}{1-\frac{\epsilon^2}{2}}$$

They then state that if $\Lambda_{01} = u$ and $\Lambda_1 = \epsilon v$ (thereby combining the theory of FROHNE [33,p10] with that of FREISE and JORDAN [23]) it follows that

$$\Lambda_{\theta,t} = \frac{1}{\delta}\left(u + \epsilon v \cos\alpha\right)$$

or
$$\Lambda_{\theta,t} = \Lambda_0\left(u + v\epsilon\cos\alpha\right)$$

which is the same as the expression given by FREISE and JORDAN. [23]

HELLER and HAMATA [26,p73] also provided a graph showing the quantities u and v plotted as functions of the eccentricity . On this graph the values of u agree with those given by FREISE and JORDAN, but the values of v are altogether different from those given by FREISE and JORDAN. [23]

HELLER and HAMATA's graph [26,p73] states in effect that for $\epsilon = 0$ we have $v = 0$ (instead of $v = 1$, as stated by FREISE and JORDAN). [23]

Now, considering HELLER and HAMATA's own statements [26,p73], $\Lambda_1 = \epsilon v$

Therefore $v = \frac{\Lambda_1}{\epsilon} \doteq \frac{1}{\epsilon}\,\frac{\epsilon}{1-\frac{\epsilon^2}{2}} = \frac{1}{1-\frac{\epsilon^2}{2}}$,

so for $\epsilon = 0$ we have $v = 1$

Therefore, HELLER and HAMATA's graph [26,p73] for v cannot be correct.

RAI [33,p52] states that for higher values of eccentricity higher orders of permeance can be considered, but that sufficient accuracy for practical purposes can be achieved by the neglect of higher combinations. He thus represents the simplified permeance wave as [33,p52]

$$\Lambda_{\theta,t} = \Lambda_0\left(1 + \epsilon\cos\alpha\right) = \Lambda_0\left\{1 + \epsilon\cos\left(\theta - \omega_\epsilon t - \phi_\epsilon\right)\right\}$$

This expression will be used in the writer's analysis.

2.13 Flux-density waves:

$$B_{\theta,t} = mmf_{\theta,t} \times \Lambda_{\theta,t}$$

$$= M_1 \cos(p\theta - \omega t - \phi) \times \Lambda_0 \{1 + \epsilon \cos(\theta - \omega_\epsilon t - \phi_\epsilon)\}$$

$$= M_1 \Lambda_0 \{\cos(p\theta - \omega t - \phi) + \epsilon \cos(p\theta - \omega t - \phi) \cos(\theta - \omega_\epsilon t - \phi_\epsilon)\}$$

$$= B_p \Big[\cos(p\theta - \omega t - \phi) + \frac{\epsilon}{2}\{\cos(p+1)\theta - (\omega + \omega_\epsilon) - (\phi + \phi_\epsilon)\} + \frac{\epsilon}{2}\{\cos(p-1)\theta - (\omega - \omega_\epsilon) - (\phi - \phi_\epsilon)\}\Big]$$

as found by RAI (1973).[33, p 53]

From the writer's foregoing discussions it is clear that these expressions are the result of considerable simplification in assumptions and that the true situation is more complex.

The term $B_p \cos(p\theta - \omega t - \phi)$ is called the "fundamental field"[33, pp 53-54] and, having the same number of pole-pairs p as the motor itself it is also called the "working field". Note that it cannot be called the "fundamental harmonic" unless $p = 1$

The remaining two terms in the above expression for $B_{\theta,t}$ are called the "eccentricity fields",[33, pp 53-54] having $p \pm 1$ pole-pairs.

The eccentricity fields increase linearly with eccentricity, but if higher powers of ϵ and higher powers of the permeance wave $[\Lambda_{01}, \Lambda_1, \Lambda_2 \ldots]$ are taken into account the eccentricity fields are more than proportional to eccentricity.

The expressions derived by FREISE and JORDAN (1962)[23] for the flux density, in their symbol nomenclature, are as follows:-

for $p = 1$

$$b(x,t) = -B_1 u \left\{ \left[1-\left(\frac{\varepsilon v}{2u}\right)^2\right] 0 \cos\left[x-\omega t-\phi'\right] \right.$$

$$+\left(\frac{\varepsilon v}{2u}\right) \cdot w_1 \cos\left[2x-(\omega+\omega_\varepsilon)t-(\phi'+\phi_\varepsilon)\right.$$

$$\left. -\left(\frac{\varepsilon v}{2u}\right)^2 \cdot w_2 \cos\left[x-2(\omega_\varepsilon-\omega)t-(2\phi_\varepsilon-\phi')\right]\right\}$$

for $p \neq 1$

$$b(x,t) = -B_1 u \left\{ 0 \cos\left[px-\omega t-\phi'\right] \right.$$

$$+\left(\frac{\varepsilon v}{2u}\right) \cdot w_1 \cos\left[(p+1)x-(\omega+\omega_\varepsilon)t-(\phi+\phi_\varepsilon)\right]$$

$$\left. +\left(\frac{\varepsilon v}{2u}\right) \cdot w_2 \cos\left[(p-1)x-(\omega-\omega_\varepsilon)t-(\phi-\phi_\varepsilon)\right]\right\}$$

where

$$b(x,t) = B_{\theta,t}$$

$$\varepsilon = \epsilon$$

$$x = \theta$$

$$\phi' = \phi$$

$$B_1 = -\frac{B_p}{u}$$

u and v have been defined and discussed in the section "Permeance wave".

0 = Factor for the reduction of the fundamental field during the starting period due to primary reactances.

$0 \approx 0{,}5$ during starting.

$0 \approx 1{,}0$ at synchronous speed.

w_1 allows for the reduction of the harmonic field (p + 1) by rotor currents.

w_2 allows for the reduction of the harmonic field (p - 1) by rotor currents.

$w_1, w_2 \leqq 1{,}0$ for squirrel cage rotors.

$w_1, w_2 \approx 1{,}0$ for slipring rotors.

The expressions of FREISE [23] and JORDAN [33, p 53] are similar to those found by RAI except for more detailed attention to the coefficients by the introduction of u, v, w_1, w_2 and 0. Also for p = 1 they have considered a higher order term containing $\left(\frac{\varepsilon v}{2u}\right)^2$

HELLER and HAMATA (1977) [28, pp 71-76] repeat FREISE and JORDAN's [23] expressions except that they omit the factor 0 and put ω_1 instead of W_1, ω_2 instead of W_2 and they do not define ω_1 and ω_2

5.2.14 Radial electromagnetic force waves - a review:

FREISE and JORDAN (1962) [23] state that it is necessary to take the square of the magnetic field strength to determine the unbalanced magnetic pull.

They state:-

for p = 1

$$b^2(x,t) = (B_1 u)^2 O\left[1 - \left(\frac{\varepsilon v}{2u}\right)^2\right] W_1 \frac{\varepsilon v}{2u} \cos(x - \omega_\varepsilon t - \phi_\varepsilon) + \ldots$$

for p $\neq$ 1

$$b^2(x,t) = (B_1 u)^2 O\left[\frac{W_1 + W_2}{2}\right] \frac{\varepsilon v}{u} \cos(x - \omega_\varepsilon t - \phi_\varepsilon) + \ldots$$

Without any further proof they go on to state the following expressions, for the unbalanced magnetic pull, F_1 :-

for p = 1

$$F_1 = \pi R l_e \frac{(B_1 u)^2}{2\mu_0}\left[1 - \left(\frac{\varepsilon v}{2u}\right)^2\right] \frac{\varepsilon v}{2u} \cdot O\, W_1$$

for p $\neq$ 1

$$F_1 = \pi R l_e \frac{(B_1 u)^2}{2\mu_0} \frac{\varepsilon v}{u} \cdot O \cdot \left(\frac{W_1 + W_2}{2}\right)$$

where

l_e = effective length of core

and other symbols have been defined before.

They make no further investigation or comment on the radial electro magnetic forces.

RAI (1973) [33, p 68] proceeds to calculate the unbalanced magnetic pull from the expression.

$$F_{\theta,t} = \frac{B^2_{\theta,t}}{2\mu_0}$$

Then he goes on to state [33, p69] that by substituting his value of $B_{\theta,t}$ (refer earlier section "Flux-density waves") one has:-

for $p \neq 1$,

$$F_{\theta,t} = \frac{B_p\,(B_{p+1} + B_{p-1})}{2\mu_0} \cos(\theta - \omega_\varepsilon t - \phi_\varepsilon)$$

and for p = 1 where homopolar flux is negligible,

$$F_{\theta,t} = \frac{B_p \times B_{p+1}}{2\mu_0} \cos(\theta - \omega_\varepsilon t - \phi_\varepsilon)$$

Now, these statements are incorrect because:-

1. the expressions given for $F_{\theta,t}$ do not follow from substituting $B_{\theta,t}$. In fact a proper substitution of $B_{\theta,t}$ would yield a more complex expression as will be shown presently.

   The portions of the expressions $B_p\,(B_{p+1} + B_{p-1})$ for $p \neq 1$ and $B_p \times B_{p+1}$ for p = 1 (homopolar flux negligible) result from selecting only those fields differing by one pole-pair. On the basis of FREISE and JORDAN's work [23] (1962) it was accepted by VON KAEHNE (1963) that the [49, p13] interaction of two fields differing by one pole-pair [33, p69] would produce unbalanced magnetic pull. RAI has accepted and used this result but the derivation and proof resulting from a proper substitution of $B_{\theta,t}$ is more involved and will be studied later in this thesis.

2. The portion of the expression $\cos(\theta - \omega_\varepsilon t - \phi_\varepsilon)$ does not follow from substituting $B_{\theta,t}$. The factor $\cos(\theta - \omega_\varepsilon t - \phi_\varepsilon)$ has the function of resolving $F_{\theta,t}$ in the direction of the line of eccentricity, thus

$$F_{\theta,t} = \frac{B^2_{\theta,t}}{2\mu_0}$$

   Therefore,

$$F_{\theta,t} \cos(\theta - \omega_\varepsilon t - \phi_\varepsilon) = \frac{B^2_{\theta,t}}{2\mu_0} \cos(\theta - \omega_\varepsilon t - \phi_\varepsilon)$$

   and not

$$F_{\theta,t} = \frac{B^2_{\theta,t}}{2\mu_0} \cos(\theta - \omega_\varepsilon t - \phi_\varepsilon)$$

and also not $$F_{\theta,t} = \frac{B_p(B_{p+1} + B_{p-1})}{2\mu_0} \cos(\theta - \omega_\varepsilon t - \phi_\varepsilon)$$

and also not $$F_{\theta,t} = \frac{B_p \times B_{p+1}}{2\mu_0} \cos(\theta - \omega_\varepsilon t - \phi_\varepsilon)$$

[33, p69]
RAI then goes on to state that the unbalanced magnetic pull,

$$F_u = \frac{lr}{2\mu_0} \int_0^{2\pi} F_{\theta,t} \cos(\theta - \phi_\varepsilon)$$

This expression is also incorrect.

The correct expression is $$F_u = \frac{lr}{2\mu_0} \int_0^{2\pi} B^2_{\theta,t} \cos(\theta - \phi_\varepsilon)\, d\theta$$

and $$F_u = lr \int_0^{2\pi} F_{\theta,t} \cos(\theta - \phi_\varepsilon)\, d\theta$$

where l = rotor length
r = radius of stator bore.

Of course, if RAI's incorrect expressions for $F_{\theta,t}$ were substituted into the correct expression for F_u they would still give the incorrect answers.

[33, pp 69-70]
RAI then derives expressions for the unbalanced magnetic pull. Having omitted the factors u, V, o, W_1 and W_2 from his expressions for $B_{\theta,t}$ these factors do not appear in his expressions for u.m.p. His expressions for u.m.p. are fundamentally incorrect as will be explained in a later section ("To find the unbalanced magnetic pull"). As in FREISE and JORDAN'S [23] expressions for u.m.p., RAI cal- [33, p70] culates the u.m.p. for p $\neq$ 1 to be twice the value of the u.m.p. for p = 1, where homopolar flux is negligible. This is obtained by the simple device of omitting the term B_{p-1} from the expression, but proper proof is not given, nor is it shown what would happen if p = 1 and homopolar flux was not negligible.

[26, p75] [23]
HELLER and HAMATA (1977) repeat FREISE and JORDAN'S expressions for unbalanced magnetic pull except that they omit the factor 0 and put ω_1 instead of W_1, ω_2 instead of W_2 and they do not define ω_1 and ω_2.

[26, p75]
These expressions are given without proof but preceeded by the statement that the "one-sided magnetic pull between the stator and the rotor is conditioned by a two-pole magnetic pressure wave appearing whenever two fields having a pole-pair difference equal to 1 exist in the airgap. In such a case, the amplitude f_e of the one-sided magnetic pull is proportional to the product of the two induction waves having a pole-pair difference of 1". There is no mention of any other forces present with an eccentrically supported rotor.

[49, pp 7, 13]

VON KAEHNE (1963) states that the interaction between fields having differing numbers of pole-pairs causes force distributions which result in different deformations of the rotor and of the stator. The order r of the force wave defines the deformation. This order r is given as the sum or the difference of the pole-pair numbers p_1 and p_2 of two harmonic fields which produce the force wave and the deformation. [49, p7]

$$r = p_1 \pm p_2$$

[49, p13]

Thus, if B_p = Fundamental field with p pole-pairs.

$B_{\epsilon 1} = B_{p+1}$ = First harmonic field with (p + 1) pole-pairs.

$B_{\epsilon 2} = B_{p-1}$ = Second harmonic field with (p-1) pole-pairs.

Then, these three fields interact and give the following force waves of the order r:

$$\left.\begin{array}{l} B_p - B_{\epsilon 1} \\ \\ B_p - B_{\epsilon 2} \end{array}\right\} |r| = 1$$

Two different unbalanced magnetic pulls which must be added.

$$B_{\epsilon 1} - B_{\epsilon 2}\ ;\ r = 2$$

Deformation of stator and rotor to an elliptical shape.

$$\left.\begin{array}{ll} B_p + B_{\epsilon 2}\ ; & r = 2p-1 \\ B_{\epsilon 1} + B_{\epsilon 2}\ ; & r = 2p \\ B_p + B_{\epsilon 1}\ ; & r = 2p+1 \end{array}\right\} \quad \text{Deformations to polygons with} \left\{\begin{array}{ll} 2(2p-1) & \text{nodes} \\ 2.2p & \text{nodes} \\ 2(2p+1) & \text{nodes} \end{array}\right.$$

[49, p13]

VON KAEHNE then goes on to state that a two-pole machine must be treated separately because the field $B_{\epsilon 2}$ is homopolar and does not contribute to unbalanced magnetic pull. This statement is not really correct. In the case of a two-pole machine it is true that $B_{\epsilon 2}$ is homopolar. If the homopolar flux is present, that is, if the machine is constructed in such a way that the reluctance to homopolar flux is low then $B_{\epsilon 2}$ will, in fact, contribute to unbalanced magnetic pull. However, if the reluctance to the homopolar flux is high so that the homopolar flux is negligible then $B_{\epsilon 2}$ will make no significant contribution to the u.m.p. Then, as VON KAEHNE states, [49, p13]

$$B_p - B_{\epsilon 1} = |r| = 1$$

Only one unbalanced magnetic pull exists.

$$B_p + B_{\epsilon 1} \qquad r = 3$$

Deformation into a triangle with six nodes.

VON KAEHNE [49, p13] concludes that eccentricity causes not only unbalanced magnetic pull but also deformations of the stator and rotor giving rise to vibration and noise.

Mention of RAI's [33] work on vibratory forces has already been made earlier in this thesis.

5.2.15 Radial electromagnetic force waves - mathematical development:

The writer will now proceed to develop the subject mathematically.

As found earlier,

$$B_{\theta,t} = B_p \Big[\cos(p\theta - \omega t - \phi) + \frac{\epsilon}{2}\cos\{(p+1)\theta - (\omega + \omega_\epsilon)t - (\phi + \phi_\epsilon)\} + \frac{\epsilon}{2}\cos\{(p-1)\theta - (\omega - \omega_\epsilon)t - (\phi - \phi_\epsilon)\}\Big]$$

$$= \bar{B}_p + \bar{B}_{p+1} + \bar{B}_{p-1}$$

where,

$$\bar{B}_p = B_p \cos(p\theta - \omega t - \phi)$$

$$\bar{B}_{p+1} = \frac{\epsilon}{2} B_p \cos\{(p+1)\theta - (\omega + \omega_\epsilon)t - (\phi + \phi_\epsilon)\}$$

$$\bar{B}_{p-1} = \frac{\epsilon}{2} B_p \cos\{(p-1)\theta - (\omega - \omega_\epsilon)t - (\phi - \phi_\epsilon)\}$$

$$B^2_{\theta,t} = (\bar{B}_p + \bar{B}_{p+1} + \bar{B}_{p-1})^2$$

$$= \bar{B}_p^2 + \bar{B}_p\bar{B}_{p+1} + \bar{B}_p\bar{B}_{p-1} + \bar{B}_{p+1}\bar{B}_p + \bar{B}_{p+1}^2 + \bar{B}_{p+1}\bar{B}_{p-1} + \bar{B}_{p-1}\bar{B}_p + \bar{B}_{p-1}\bar{B}_{p+1} + \bar{B}_{p-1}^2$$

$$= \underset{I}{\bar{B}_p^2} + \underset{II}{2\bar{B}_p\bar{B}_{p+1}} + \underset{III}{2\bar{B}_p\bar{B}_{p-1}} + \underset{IV}{\bar{B}_{p+1}^2} + \underset{V}{2\bar{B}_{p+1}\bar{B}_{p-1}} + \underset{VI}{\bar{B}_{p-1}^2}$$

Each term of this expression will now be considered separately.

$$\text{I} \quad \bar{B}_p^2 = \left[B_p \cos(p\theta - \omega t - \phi) \right]^2$$
$$= B_p^2 \cos^2(p\theta - \omega t - \phi)$$
$$= B_p^2 \frac{\{1 + \cos 2(p\theta - \omega t - \phi)\}}{2} \qquad \left[\cos^2 A = \tfrac{1}{2} + \tfrac{1}{2} \cos 2A \right]$$
$$= B_p^2 \left\{ \tfrac{1}{2} + \tfrac{1}{2} \cos(2p\theta - 2\omega t - 2\phi) \right\}$$

$$\text{II} \quad 2\bar{B}_p \bar{B}_{p+1} = 2B_p \cos(p\theta - \omega t - \phi) \cdot \tfrac{\epsilon}{2} B_p \cos\left\{ (p+1)\theta - (\omega + \omega_\epsilon)t - (\phi + \phi_\epsilon) \right\}$$
$$= \epsilon B_p^2 \cos(p\theta - \omega t - \phi) \cos\left\{ (p+1)\theta - (\omega + \omega_\epsilon)t - (\phi + \phi_\epsilon) \right\}$$
$$= \epsilon B_p^2 \cdot \tfrac{1}{2} \Big[\cos\left\{ p\theta - \omega t - \phi + (p+1)\theta - (\omega + \omega_\epsilon)t - (\phi + \phi_\epsilon) \right\}$$
$$+ \cos\left\{ p\theta - \omega t - \phi - (p+1)\theta + (\omega + \omega_\epsilon)t + (\phi + \phi_\epsilon) \right\} \Big]$$
$$\left[\cos A \cos B = \tfrac{1}{2} \cos(A+B) + \tfrac{1}{2} \cos(A-B) \right]$$
$$= \tfrac{\epsilon}{2} B_p^2 \Big[\cos\left\{ (2p+1)\theta - (2\omega + \omega_\epsilon)t - 2(\phi + \phi_\epsilon) \right\}$$
$$+ \cos\left\{ -\theta + \omega_\epsilon t + \phi_\epsilon \right\} \Big]$$
$$= \tfrac{\epsilon}{2} B_p^2 \Big[\cos\left\{ (2p+1)\theta - (2\omega + \omega_\epsilon)t - (2\phi + \phi_\epsilon) \right\} + \cos(\theta - \omega_\epsilon t - \phi_\epsilon) \Big]$$

$$\text{III} \quad 2\bar{B}_p \bar{B}_{p-1} = 2B_p \cos(p\theta - \omega t - \phi) \cdot \tfrac{1}{2} \epsilon B_p \cos\{(p-1)\theta - (\omega - \omega_\epsilon)t - (\phi - \phi_\epsilon)\}$$
$$= \epsilon B_p^2 \cos(p\theta - \omega t - \phi) \cos\{(p-1)\theta - (\omega - \omega_\epsilon)t - (\phi - \phi_\epsilon)\}$$
$$= \tfrac{\epsilon}{2} B_p^2 \left[\cos\left\{ (2p-1)\theta - (2\omega - \omega_\epsilon)t - (2\phi - \phi_\epsilon) \right\} + \cos(\theta - \omega_\epsilon t - \phi_\epsilon) \right]$$
$$\left[\cos A \cos B = \tfrac{1}{2} \cos(A+B) + \tfrac{1}{2} \cos(A-B) \right]$$

$$\underline{IV} \quad \bar{B}_{p+1}^2 = \left[\tfrac{1}{2} \epsilon B_p \cos\left\{ (p+1)\theta - (\omega + \omega_\epsilon)t - (\phi + \phi_\epsilon) \right\} \right]^2$$

$$= \frac{\epsilon^2}{4} B_p^2 \cos^2\left\{ (p+1)\theta - (\omega + \omega_\epsilon)t - (\phi + \phi_\epsilon) \right\}$$

$$= \frac{\epsilon^2}{8} B_p^2 \left[1 + \cos\left\{ 2(p+1)\theta - 2(\omega + \omega_\epsilon)t - 2(\phi + \phi_\epsilon) \right\} \right]$$

$$\left[\cos^2 A = \tfrac{1}{2} + \tfrac{1}{2} \cos 2A \right]$$

$$\underline{V} \quad 2\bar{B}_{p+1}\bar{B}_{p-1} = 2\tfrac{\epsilon}{2} B_p \cos\left\{ (p+1)\theta - (\omega + \omega_\epsilon)t - (\phi + \phi_\epsilon) \right\} \cdot \tfrac{\epsilon}{2} B_p \cos\left\{ (p-1)\theta - (\omega - \omega_\epsilon)t - (\phi - \phi_\epsilon) \right\}$$

$$= \frac{\epsilon^2}{2} B_p^2 \cos\left\{ (p+1)\theta - (\omega + \omega_\epsilon)t - (\phi + \phi_\epsilon) \right\} \cos\left\{ (p-1)\theta - (\omega - \omega_\epsilon)t - (\phi - \phi_\epsilon) \right\}$$

$$= \frac{\epsilon^2}{4} B_p^2 \left\{ \cos(2p\theta - 2\omega t - 2\phi) + \cos(2\theta - 2\omega_\epsilon t - 2\phi_\epsilon) \right\}$$

$$\left[\cos A \cos B = \tfrac{1}{2} \cos(A+B) + \tfrac{1}{2} \cos(A-B) \right]$$

$$\underline{VI} \quad \bar{B}_{p-1}^2 = \left[\tfrac{\epsilon}{2} B_p \cos\left\{ (p-1)\theta - (\omega - \omega_\epsilon)t - (\phi - \phi_\epsilon) \right\} \right]^2$$

$$= \frac{\epsilon^2}{4} B_p^2 \cos^2\left\{ (p-1)\theta - (\omega - \omega_\epsilon)t - (\phi - \phi_\epsilon) \right\}$$

$$= \frac{\epsilon^2}{8} B_p^2 \left[1 + \cos\left\{ 2(p-1)\theta - 2(\omega - \omega_\epsilon)t - 2(\phi - \phi_\epsilon) \right\} \right]$$

$$\left[\cos^2 A = \tfrac{1}{2} + \tfrac{1}{2} \cos 2A \right]$$

$$B^2_{\theta,t} = B_p^2 + 2\bar{B}_p\bar{B}_{p+1} + 2\bar{B}_p B_{p-1} + \bar{B}^2_{p+1} + 2\bar{B}_{p+1}\bar{B}_{p-1} + \bar{B}^2_{p-1}$$

$$\begin{aligned}
= B_p^2 \Big[\tfrac{1}{2} \quad & + \tfrac{1}{2}\cos(2p\theta - 2\omega t - 2\phi) && \text{I} \\
+ \tfrac{\epsilon}{2}\cos\{(2p+1)\theta - (2\omega+\omega_\epsilon)t - (2\phi+\phi_\epsilon)\} & + \tfrac{\epsilon}{2}\cos(\theta - \omega_\epsilon t - \phi_\epsilon) && \text{II} \\
+ \tfrac{\epsilon}{2}\cos\{(2p+1)\theta - (2\omega-\omega_\epsilon)t - (2\phi-\phi_\epsilon)\} & + \tfrac{\epsilon}{2}\cos(\theta - \omega_\epsilon t - \phi_\epsilon) && \text{III} \\
+ \tfrac{\epsilon^2}{8} \quad & + \tfrac{\epsilon^2}{8}\cos\{2(p+1)\theta - 2(\omega+\omega_\epsilon)t - 2(\phi+\phi_\epsilon)\} && \text{IV} \\
+ \tfrac{\epsilon^2}{4}\cos(2p\theta - 2\omega t - 2\phi) & + \tfrac{\epsilon^2}{4}\cos(2\theta - 2\omega_\epsilon t - 2\phi_\epsilon) && \text{V} \\
+ \tfrac{\epsilon^2}{8} \quad & + \tfrac{\epsilon^2}{8}\cos\{2(p-1)\theta - 2(\omega-\omega_\epsilon)t - 2(\phi-\phi_\epsilon)\} \Big] && \text{VI}
\end{aligned}$$

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16 <u>To find the unbalanced magnetic pull</u>:

For the purpose of calculating the attractive forces between the rotor and the stator, $F_{\theta,t}$, the radial magnetic flux density $B_{\theta,t}$ at any given angle is assumed to be the same at the rotor surface as it is at the stator surface. Although not completely correct, this is assumed by all investigators.

The airgap is bounded by two surfaces. Ideally each of these surfaces is cylindrical. $B_{\theta,t}$ is considered to act normally to an element of cylindrical surface at the airgap. Some investigators [49][5][6] consider this element of cylindrical surface area to be part of the rotor surface and that the radius to this surface is the rotor radius. Other [26][31][23][33] investigators consider this element of cylindrical surface area to be part of the stator surface and that the radius to this surface is the stator radius. For the purpose of the writer's calculation of the unbalanced magnetic pull no distinction will be made between the stator radius and the rotor radius. These radii will be designated as r, which can be regarded as the mean airgap radius.

Let l = effective length of rotor and stator cores.
Then, the area of an element of the cylindrical surface of the rotor (or of the stator) = $l r\, d\theta$

The attractive force acting on this elemental area of the rotor (or of the stator) = $F_{\theta,t}\, l r\, d\theta = l r F_{\theta,t}\, d\theta$.

The component of this force in the direction of eccentricity

$$= l r F_{\theta,t}\, d\theta . \cos(\theta - \omega_\epsilon t - \phi_\epsilon)$$

$$= l r F_{\theta,t} \cos(\theta - \omega_\epsilon t - \phi_\epsilon)\, d\theta$$

$$= l r \frac{B^2_{\theta,t}}{2\mu_0} \cos(\theta - \omega_\epsilon t - \phi_\epsilon)\, d\theta$$

The total resultant force in the direction of eccentricity

$$F_u = \int_0^{2\pi} l r \frac{B^2_{\theta,t}}{2\mu_0} \cos(\theta - \omega_\epsilon t - \phi_\epsilon)\, d\theta$$

$$= \frac{l r}{2\mu_0} \int_0^{2\pi} B^2_{\theta,t} \cos(\theta - \omega_\epsilon t - \phi_\epsilon)\, d\theta$$

$$= \frac{l r}{2\mu_0} \int_0^{2\pi} \left[\bar{B}_p + \bar{B}_{p+1} + \bar{B}_{p-1}\right]^2 \cos(\theta - \omega_\epsilon t - \phi_\epsilon)\, d\theta$$

$$= \frac{l r}{2\mu_0} \int_0^{2\pi} \left[\bar{B}_p^2 + 2\bar{B}_p\bar{B}_{p+1} + 2\bar{B}_p\bar{B}_{p-1} + \bar{B}^2_{p+1} + 2\bar{B}_{p+1}\bar{B}_{p-1} + \bar{B}^2_{p-1}\right] \cos(\theta - \omega_\epsilon t - \phi_\epsilon)\, d\theta$$

This is the unbalanced magnetic pull.

In the preceeding section it was proved that $B^2_{\theta,t}$ comprizes constant terms and terms which are of the form $\cos(m\theta - \beta)$, where β is independent of θ.

Therefore all the terms of $B^2_{\theta,t}\cos(\theta - \omega_\epsilon t - \phi_\epsilon)$ are of the form

$A \cos(\theta - \omega_\epsilon t - \phi_\epsilon)$, where A is a constant or

$\cos(m\theta - \beta)\cos(\theta - \omega_\epsilon t - \phi_\epsilon)$

Now, $\cos(m\theta - \beta) = \cos m\theta \cos\beta + \sin m\theta \sin\beta$

$$\left[\cos(A - B) = \cos A \cos B + \sin A \sin B\right]$$

and $\cos(\theta - \omega_\epsilon t - \phi_\epsilon) = \cos(\theta - \gamma)$, where $\gamma = \omega_\epsilon t + \phi_\epsilon$

and γ is independent of θ

$$= \cos\theta \cos\gamma + \sin\theta \sin\gamma$$

Therefore, $\cos(m\theta - \beta)\cos(\theta - \omega_\epsilon t - \phi_\epsilon)$

$$= \left[\cos m\theta \cos\beta + \sin m\theta \sin\beta\right]\left[\cos\theta \cos\gamma + \sin\theta \sin\gamma\right]$$

$$= \cos\beta \cos\gamma \cos m\theta \cos\theta + \cos\beta \sin\gamma \cos m\theta \sin\theta$$
$$+ \sin\beta \cos\gamma \sin m\theta \cos\theta + \sin\beta \sin\gamma \sin m\theta \sin\theta$$

Now, seeing that β and γ are independent of θ, it follows that $\cos\beta$ $\sin\beta$, $\cos\gamma$ and $\sin\gamma$ are independent of θ
Therefore, $\cos\beta$, $\sin\beta$, $\cos\gamma$ and $\sin\gamma$ can be regarded as constants when calculating

$$\int_0^{2\pi} F_{\theta,t} \cos(\theta - \omega_\epsilon t - \phi_\epsilon)\, d\theta .$$

Therefore, for the purpose of calculating this integral all terms of $F_{\theta,t}\cos(\theta - \omega_\epsilon t - \phi_\epsilon)$ except the terms of the form $A\cos(\theta - \omega_\epsilon t - \phi_\epsilon)$ can be regarded as being of the following forms:-

$\cos m\theta \cos\theta$
$\cos m\theta \sin\theta$
$\sin m\theta \cos\theta$
$\sin m\theta \sin\theta$

The definite integral of each of these forms will now be calculated separately.

<u>$\cos m\theta \cos\theta$</u>

$$\int \cos m\theta \cos n\theta \, d\theta = \frac{\sin(m+n)\theta}{2(m+n)} + \frac{\sin(m-n)\theta}{2(m-n)} + C$$

Therefore, for $n = 1$

$$\int \cos m\theta \cos\theta \, d\theta = \frac{\sin(m+1)\theta}{2(m+1)} + \frac{\sin(m-1)}{2(m-1)} + C$$

Therefore,

$$\int_0^{2\pi} \cos m\theta \cos\theta \, d\theta = \left[\frac{\sin(m+1)\theta}{2(m+1)} + \frac{\sin(m-1)\theta}{2(m-1)}\right]_0^{2\pi}$$

$$= \left[(0+0) - (0+0)\right] \text{ for } m \neq 1 .$$

$$= 0 \text{ for } m \neq 1$$

Therefore a term of form $\cos m\theta \cos\theta$ for $m \neq 1$ will not contribute to the unbalanced magnetic pull.

However, if $m=1$, $m-1=0$

and the term $\dfrac{\sin(m-1)\theta}{2(m-1)}$ is indeterminate

Therefore, for $m=1$, the solution of $\int \cos m\theta \cos\theta \, d\theta$ is indeterminate by this method.

However, for $m=1$ $\int \cos m\theta \cos\theta \, d\theta = \int \cos^2\theta \, d\theta$

$$= \frac{\theta}{2} + \frac{1}{4}\sin 2\theta + C$$

Therefore $\displaystyle\int_0^{2\pi} \cos^2\theta \, d\theta = \left[\frac{\theta}{2} + \frac{1}{4}\sin 2\theta\right]_0^{2\pi}$

$$= \left[\left(\frac{2\pi}{2} + 0\right) - (0+0)\right]$$

$$= \pi$$

Therefore, for a term of $F_{\theta,t}\cos(\theta - \omega_\epsilon t - \phi_\epsilon)$ having the form $\cos m\theta \cos\theta$, there will be unbalanced magnetic pull for $m=1$, but no unbalanced magnetic pull for $m \neq 1$

$\underline{\cos m\theta \sin\theta}$

$$\int \cos m\theta \sin n\theta \, d\theta = \frac{-\cos(m+n)\theta}{2(m+n)} + \frac{\cos(m-n)\theta}{2(m-n)} + C$$

Therefore, for $n=1$

$$\int \cos m\theta \sin\theta \, d\theta = \frac{-\cos(m+1)\theta}{2(m+1)} + \frac{\cos(m-1)\theta}{2(m-1)} + C$$

Therefore,

$$\int_0^{2\pi} \cos m\theta \sin\theta \, d\theta = \left[\frac{-\cos(m+1)\theta}{2(m+1)} + \frac{\cos(m-1)\theta}{2(m-1)}\right]_0^{2\pi}$$

$$= \left[\left(\frac{-1}{2(m+1)} + \frac{1}{2(m-1)}\right) - \left(\frac{-1}{2(m+1)} + \frac{1}{2(m-1)}\right)\right] \text{ for } m \neq 1$$

$$= 0 \text{ for } m \neq 1$$

Therefore, a term of the form $\cos m\theta \sin\theta$ for $m \neq 1$ will not contribute to the unbalanced magnetic pull.

However, if $m=1$, $m-1=0$,

the term $\frac{\cos(m-1)\theta}{2(m-1)}$ is infinite. Subtraction of an infinite term from an infinite term renders the definite integral indeterminate. Therefore, for $m=1$, the solution of $\int_0^{2\pi} \cos m\theta \sin\theta \, d\theta$ is indeterminate by this method.

However, for $m=1$, $\int \cos\theta \sin\theta \, d\theta = \int \frac{1}{2} \sin 2\theta \, d\theta = -\frac{\cos 2\theta}{4} + C$

$$\text{and} \quad \int_0^{2\pi} \cos\theta \sin\theta \, d\theta = \left[-\frac{\cos 2\theta}{4} \right]_0^{2\pi} = 0$$

Therefore, for a term of $F_{\theta,t} \cos(\theta - \omega_\epsilon t - \phi_\epsilon)$ having the form $\cos m\theta \sin\theta$ there will be no unbalanced magnetic pull, whatever the value of m.

$\underline{\sin m\theta \cos\theta}$

$$\int \sin m\theta \cos n\theta = -\frac{\cos(m+n)\theta}{2(m+n)} - \frac{\cos(m-n)\theta}{2(m-n)} + C$$

Therefore, for $n=1$

$$\int \sin m\theta \cos\theta = -\frac{\cos(m+1)\theta}{2(m+1)} - \frac{\cos(m-1)\theta}{2(m-1)} + C$$

Therefore

$$\int_0^{2\pi} \sin m\theta \cos\theta \, d\theta = \left[-\frac{\cos(m+1)\theta}{2(m+1)} - \frac{\cos(m-1)\theta}{2(m-1)} \right]_0^{2\pi}$$

$$= \left[(-0-0) - (-0-0) \right] \text{ for } m \neq 1$$

$$= 0 \text{ for } m \neq 1$$

Therefore, a term of the form $\sin m\theta \cos\theta$ for $m \neq 1$ will not contribute to the unbalanced magnetic pull.

However, if $m=1$, $m-1=0$,

the term $\frac{\cos(m-1)}{2(m-1)}$ is infinite. Subtraction of an infinite term from an infinite term renders the definite integral indeterminate. Therefore, for $m=1$, the solution of $\int_0^{2\pi} \sin m\theta \cos\theta \, d\theta$ is indeterminate by this method.

However, for $m=1$, $\int \sin\theta \cos\theta \, d\theta = \int \frac{1}{2} \sin 2\theta \, d\theta = -\frac{\cos 2\theta}{4} + C$

$$\text{and} \quad \int_0^{2\pi} \sin\theta \cos\theta \, d\theta = \left[-\frac{\cos 2\theta}{4} \right]_0^{2\pi} = 0$$

Therefore, for a term of $F_{\theta,t}\cos(\theta-\omega_\epsilon t-\phi_\epsilon)$ having the form $\cos m\theta \sin\theta$ there will be no unbalanced magnetic pull, whatever the value of m.

$\underline{\sin m\theta \sin\theta}$

$$\int \sin m\theta \sin n\theta \, d\theta = \frac{-\sin(m+n)\theta}{2(m+n)} + \frac{\sin(m-n)\theta}{2(m-n)} + C$$

Therefore, for $n=1$

$$\int \sin m\theta \sin\theta \, d\theta = -\frac{\sin(m+1)\theta}{2(m+1)} + \frac{\sin(m-1)\theta}{2(m-1)} + C$$

Therefore,

$$\int_0^{2\pi} \sin m\theta \sin\theta \, d\theta = \left[-\frac{\sin(m+1)\theta}{2(m+1)} + \frac{\sin(m-1)\theta}{2(m-1)}\right]_0^{2\pi}$$

$$= \left[(-0+0)-(-0+0)\right] \quad \text{for } m \neq 1$$

$$= 0 \quad \text{for } m \neq 1$$

Therefore, a term of the form $\sin m\theta \sin\theta$ for $m\neq 1$ will not contribute to the unbalanced magnetic pull.

However, if $m=1$, $m-1=0$

the term $\dfrac{\sin(m-1)\theta}{2(m-1)}$ is indeterminate

Therefore, for $m=1$ the solution of $\int \sin m\theta \sin\theta \, d\theta$ is indeterminate by this method.

However, for $m=1$ $\int \sin^2\theta \, d\theta = \frac{\theta}{2} - \frac{1}{4}\sin 2\theta + C$

$$\text{and} \quad \int_0^{2\pi} \sin^2\theta \, d\theta = \left[\frac{\theta}{2} - \frac{1}{4}\sin 2\theta\right]_0^{2\pi}$$

$$= \left[\left(\frac{2\pi}{2}-0\right)-(0-0)\right]$$

$$= \pi$$

Therefore, for a term of $F_{\theta,t}\cos(\theta-\omega_\epsilon t-\phi_\epsilon)$ having the form $\sin m\theta \sin\theta$ there will be an unbalanced magnetic pull for $m=1$, but no unbalanced magnetic pull for $m\neq 1$

Constant terms of $B^2_{\theta,t}$

The constant terms of $B^2_{\theta t}$ appear in the expression $B^2_{\theta,t}\cos(\theta - \omega_\epsilon t - \phi_\epsilon)$ in the form $A\cos(\theta - \omega'_\epsilon t - \phi_\epsilon)$, where $A = \frac{1}{2}$ or $A = \frac{\epsilon^2}{8}$

$$\int_0^{2\pi} A\cos(\theta - \omega_\epsilon t - \phi_\epsilon)\,d\theta = \int_0^{2\pi} A\cos(\theta - \gamma)\,d\theta$$

$$= A\int_0^{2\pi}[\cos\theta\cos\gamma + \sin\theta\sin\gamma]\,d\theta$$

$$= A\cos\gamma\int_0^{2\pi}\cos\theta\,d\theta + A\sin\gamma\int_0^{2\pi}\sin\theta\,d\theta$$

$$= A\cos\gamma\left[\sin\theta\right]_0^{2\pi} + A\sin\gamma\left[-\cos\theta\right]_0^{2\pi}$$

$$= A\cos\gamma[0-0] + A\sin\gamma[(-1)-(-1)]$$

$$= 0$$

Therefore, these constant terms of $B^2_{\theta,t}$ produce no unbalanced magentic pull.

$$\int_0^{2\pi} B^2_{\theta,t}\cos(\theta - \omega_\epsilon t - \phi_\epsilon)\,d\theta$$

Now, having considered all the terms of $B^2_{\theta,t}\cos(\theta - \omega_\epsilon t - \phi_\epsilon)$ it has been shown that the only terms which contribute to the unbalanced magnetic pull are the terms of the form $\cos(m\theta - \beta)\cos(\theta - \omega_\epsilon t - \phi_\epsilon)$ where $m = 1$, that is, $\cos(\theta - \beta)\cos(\theta - \omega_\epsilon t - \phi_\epsilon)$. There are only two terms of $B^2_{\theta,t}\cos(\theta - \omega_\epsilon t - \phi_\epsilon)$ which have this form for all values of p and these two terms are <u>each</u> equal to

$$\frac{\epsilon}{2}B_p^2\cos(\theta - \omega_\epsilon t - \phi_\epsilon)\cos(\theta - \omega_\epsilon t - \phi_\epsilon)$$

Now considering each of these terms and taking the definite integral we have

$$\frac{\epsilon}{2}B_p^2\int_0^{2\pi}\cos(\theta - \omega_\epsilon t - \phi_\epsilon)\cos(\theta - \omega_\epsilon t - \phi_\epsilon)\,d\theta$$

This expression may be rewritten in several ways as follows:-

$$\frac{\epsilon}{2} B_p^2 \int_0^{2\pi} \cos^2(\theta - \omega_\epsilon t - \phi_\epsilon)\, d\theta$$

or $$\frac{\epsilon}{2} B_p^2 \int_0^{2\pi} \cos(\theta-\beta)\cos(\theta-\gamma)\, d\theta$$

or $$\frac{\epsilon}{2} B_p^2 \int_0^{2\pi} \cos(\theta-\beta)\cos(\theta - \omega_\epsilon t - \phi_\epsilon)\, d\theta$$

in the case of each of these two terms $\beta = \gamma = \omega_\epsilon t + \phi_\epsilon$

Now, we have already shown that

$$\cos(m\theta-\beta)\cos(\theta-\omega_\epsilon t-\phi_\epsilon) = \cos\beta\cos\gamma\cos m\theta\cos\theta + \cos\beta\sin\gamma\cos m\theta\sin\theta + \sin\beta\cos\gamma\sin m\theta\cos\theta + \sin\beta\sin\gamma\sin m\theta\sin\theta$$

Therefore, for $m=1$, we have

$$\begin{aligned}\int_0^{2\pi} \cos(\theta-\beta)\cos(\theta-\omega_\epsilon t-\phi_\epsilon)\, d\theta &= \cos\beta\cos\gamma\int_0^{2\pi}\cos^2\theta\, d\theta + \sin\beta\sin\gamma\int_0^{2\pi}\sin^2\theta\, d\theta \\ &= \cos\beta\cos\gamma\,[\pi] + \sin\beta\sin\gamma\,[\pi] \\ &= \pi(\cos\beta\cos\gamma + \sin\beta\sin\gamma) \\ &= \pi(\cos^2\beta + \sin^2\beta) \\ &= \pi\end{aligned}$$

Now this is independent of the value of $\omega_\epsilon t + \phi_\epsilon (=\beta=\gamma)$, that is, the contribution of each of these two terms to the unbalanced magnetic pull is independent of whether the eccentricity is stationary or rotating.

The contribution by each of the abovementioned two terms of $B^2_{\theta,t}\cos(\theta-\omega_\epsilon t-\phi_\epsilon)$ to the unbalanced magnetic pull is

$$= \frac{lr}{2\mu_0}\frac{\epsilon}{2}B_p^2\int_0^{2\pi}\cos(\theta-\omega_\epsilon t-\phi_\epsilon)\cos(\theta-\omega_\epsilon t-\phi_\epsilon)\,d\theta$$

$$= \frac{lr}{2\mu_0}\frac{\epsilon}{2}B_p^2\pi$$

$$= \frac{\pi r l B_p^2}{4\mu_0}\epsilon$$

As will be explained presently in the section "Homopolar fields", this will be the value of the unbalanced magnetic pull for p = 1 and no homopolar flux present.

If both of the abovementioned terms of $B^2_{\theta,t}\cos(\theta-\omega_\epsilon t-\phi_\epsilon)$ are present, the total unbalanced magnetic pull will be

$$= \frac{\pi r l B_p^2}{4\mu_0}\epsilon \times 2$$

$$= \frac{\pi r l B_p^2\epsilon}{2\mu_0}$$

As will be explained presently in the section "Homopolar fields", this will be the value of the unbalanced magnetic pull for $p \neq 1$

It will also be explained presently in the section "Homopolar fields" that the above values of unbalanced magnetic pull do not apply to the case where p = 1 and significant homopolar flux is present. The unbalanced magnetic pull can be stronger than the values given in the above expressions if p = 1 and significant homopolar flux is present.

Comparing the above expressions for the unbalanced magnetic pull with the expressions of FREISE and JORDAN [23] and taking the value of each of the factors u, v, 0, w_1, and w_2 as unity in FREISE and JORDAN's [23] expressions for u.m.p., it will be seen that the writer's expressions are identical to the expressions of FREISE and JORDAN. Likewise, the writer's values of u.m.p. are identical to those given by MEILER, SPERLING and [31] TIKVICKI if we take $W_1 = W_2 = 1$ in the latter's expressions. [33, p70]

However, RAI's values of unbalanced magnetic pull are different.

For $p \neq 1$ he states that the u.m.p., $F_{u1} = \dfrac{\pi l r B_p^2\epsilon}{\mu_0}$

and that for p = 1 where homopolar flux is negligible

$$F_{u2} = \frac{\pi l r B_p^2\epsilon}{2\mu_0}$$

These results are twice the magnitude of the values found by the writer.

[33, pp 69-70]
RAI, having simplified his (incorrect) expression

$$F_u = \frac{lr}{2\mu_0} \int_0^{2\pi} F_{\theta,t} \cos(\theta - \phi_\epsilon)$$

obtains $$F_{u1} = \frac{\pi l r}{2\mu_0} B_p (B_{p+1} + B_{p-1})$$

and $$F_{u2} = \frac{\pi l r}{2\mu_0} B_p \times B_{p+1}$$

Substituting the values of eccentricity fields he states that

$$F_{u1} = \frac{\pi l r B_p^2}{2\mu_0} \left\{ F(\epsilon)_{p+1} + F(\epsilon)_{p-1} \right\}$$

and that $$F_{u2} = \frac{\pi l r B_p^2}{2\mu_0} F(\epsilon)_{p+1}$$

where $$B_{p\pm 1} = B_p F(\epsilon)_{p\pm 1}$$

and $$B_{p\pm 1} = B_p \frac{\Lambda_1}{2\Lambda_{01}} = B_p \frac{\epsilon}{2} \frac{\Lambda_1/\epsilon}{\Lambda_{01}}$$

He gives the variation of $\frac{\Lambda_1}{\epsilon}$ and Λ_{01} with eccentricity in his Fig. 2. These are each slightly in excess of unity for values of eccentricity up to about 0,3. For an eccentricity of 0,3 the ratio of these is only 1,02 and, for 0,4 eccentricity, the ratio is 1,05.

Therefore, taking $$\frac{\Lambda_1/\epsilon}{\Lambda_{01}} \doteqdot 1$$

we have $B_{p\pm 1} \doteqdot B_p \frac{\epsilon}{2}$ and $B_{p\pm 1} \doteqdot B_p F(\epsilon)_{p\pm 1}$

Therefore $$F(\epsilon)_{p\pm 1} \doteqdot \frac{\epsilon}{2}$$

However, he states that for values of eccentricity smaller than say 0,3, $F(\epsilon)_{p+1}$ and $F(\epsilon)_{p-1}$ are both almost equal to the eccentricity. The writer cannot agree with this statement. On the basis of this (incorrect) assumption RAI obtains the values of F_{u1} and F_{u2} quoted above.

[5][6]
BRADFORD (1968) obtained the expression

u.m.p., $F = \dfrac{\pi D \ell B_p^2 \epsilon}{4\mu_0}$ where $D = 2r$

From his expression it follows that

$$F = \frac{\pi . 2r \ell B_p^2 \epsilon}{4\mu_0}$$

$$= \frac{\pi r \ell B_p^2 \epsilon}{2\mu_0}$$

BRADFORD does not distinguish between the cases where p = 1 and $p \neq 1$ in the derivation of his formula for u.m.p. This is because his approach is "conventional" rather than on the basis of the harmonic field theory. BRADFORD's work was applicable to a 6 - pole induction motor. Therefore his formula is presumably applicable to the case $p \neq 1$ This being accepted it will be seen that the writer's value of u.m.p. for $p \neq 1$ is identical to BRADFORD's.

BRADFORD calculated the effect on the u.m.p. of slotting of the stator and the rotor cores and of saturation of the iron. [33, pp 70-74]

RAI gave consideration to these factors as well as the effect of damper windings and other parallel paths in the stator or rotor windings. He derived correction factors for these effects as applicable to his basic formula for u.m.p. In view of RAI's incorrect basic formula for u.m.p., a deeper study of RAI's correction factors is indicated. This is beyond the scope of this thesis and would be the subject of a more advanced study.

2.17 Homopolar fields :

The radial magnetic flux in the airgap completes its path through the magnetic material included in the construction of the motor.

Normally the magnetic flux path is confined to the airgap, stator core and rotor core. However, under certain conditions the radial magnetic flux in the airgap may complete its path through other magnetic material.

For a 2-pole motor, p = 1, that is p-1=0. Under this condition the harmonic magnetic field $\bar{B}_{p-1}$, having (p-1) pole-pairs, will in fact have zero pole pairs. This field will be homopolar, that is, this field will have no pole pairs around the periphery of the airgap.

This field $\bar{B}_{p-1}$, will comprize a flux whose path is through the airgap, stator core, stator frame, endshield and/or bedplate, bearings, shaft, rotor spider (if present), and rotor core, back to the airgap.

The reluctance (or permeance) of this path will be the reluctance (or permeance) to homopolar flux. The permeance to homopolar flux clearly depends on factors in addition to the airgap permeance. If the reluctance to homopolar flux is high the homopolar flux will be negligible. In this case the term $\bar{B}_{p-1} = 0$

Now considering

$$\underset{\mathrm{I}}{B^2_{\theta,t} = \bar{B}^2_p} + \underset{\mathrm{II}}{2\bar{B}_p\bar{B}_{p+1}} + \underset{\mathrm{III}}{2\bar{B}_p\bar{B}_{p-1}} + \underset{\mathrm{IV}}{\bar{B}^2_{p+1}} + \underset{\mathrm{V}}{2\bar{B}_{p+1}\bar{B}_{p-1}} + \underset{\mathrm{VI}}{\bar{B}^2_{p-1}}$$

for $\bar{B}_{p-1} = 0$ we have:-

$$\underset{\mathrm{I}}{B^2_{\theta,t} = \bar{B}^2_p} + \underset{\mathrm{II}}{2\bar{B}_p\bar{B}_{p+1}} + \underset{\mathrm{IV}}{\bar{B}^2_{p+1}} = \left(\bar{B}_p + \bar{B}_{p+1}\right)^2$$

The term $\mathrm{III} = 2\bar{B}_p\bar{B}_{p-1} = 0$

The term $\mathrm{V} = 2\bar{B}_{p+1}\bar{B}_{p-1} = 0$

The term $\mathrm{VI} = \bar{B}^2_{p-1} = 0$

It must be emphasized that these terms disappear for p = 1 provided there is negligible homopolar flux, but must be taken into account if significant homopolar flux is present even if p = 1.

Considering $B^2_{\theta,t}$ for negligible homopolar flux under the condition p = 1, we have

$$B^2_{\theta,t} = B^2_p\Big[\tfrac{1}{2} \qquad + \tfrac{1}{2}\cos\left(2p\theta - 2\omega t - 2\phi\right) \qquad \mathrm{I}$$

$$+ \tfrac{\epsilon}{2}\cos\left\{(2p+1)\theta - (2\omega+\omega_\epsilon)t - (2\phi+\phi_\epsilon)\right\} + \tfrac{\epsilon}{2}\cos\left(\theta - \omega_\epsilon t - \phi_\epsilon\right) \qquad \mathrm{II}$$

$$+ \tfrac{\epsilon^2}{8} \qquad + \tfrac{\epsilon^2}{8}\cos\left\{2(p+1)\theta - 2(\omega+\omega_\epsilon)t - 2(\phi+\phi_\epsilon)\right\}\Big] \qquad \mathrm{IV}$$

Earlier it was shown that there were two terms which could contribute to the u.m.p. It can now be seen that, for negligible homopolar flux under the condition p = 1, one of the two terms contributing to the u.m.p. has disappeared leaving the remaining term alone to contribute to u.m.p.

Specifically the u.m.p. contributing term of $B^2_{\theta,t}$ which has disappeared is the term $\frac{\epsilon}{2} B_p^2 \cos(\theta - \omega_\epsilon t - \phi_\epsilon)$ forming part of the term $2\bar{B}_p \bar{B}_{p-1}$ (*Term III*)

The u.m.p. contributing term of $B^2_{\theta,t}$ which remains is the term $\frac{\epsilon}{2} B_p^2 \cos(\theta - \omega_\epsilon t - \phi_\epsilon)$ forming part of the term $2\bar{B}_p \bar{B}_{p+1}$ (*Term II*)

If homopolar flux is present then both the terms $\frac{\epsilon}{2} B_p^2 \cos(\theta - \omega_\epsilon t - \phi_\epsilon)$ (in terms *II* and *III*) will contribute to the unbalanced magnetic pull. In addition the term $\frac{\epsilon}{2} B_p^2 \cos\{(2p-1)\theta - (2\omega - \omega_\epsilon) - (2\phi - \phi_\epsilon)\}$ (*in Term III*) will, for p = 1, have the value $\frac{\epsilon}{2} B_p^2 \cos\{\theta - (2\omega - \omega_\epsilon) - (2\phi - \phi_\epsilon)\}$ and therefore, being of the form $\cos(m\theta - \beta)$, where m = 1, will also contribute to the u.m.p. Therefore if the reluctance to homopolar flux is very low, the u.m.p. for p = 1 can be greater than the u.m.p. for the condition $p \neq 1$.

Now, $$\int_0^{2\pi} \cos\{\theta - (2\omega - \omega_\epsilon) - (2\phi - \phi_\epsilon)\} \cos(\theta - \omega_\epsilon t - \phi_\epsilon)\, d\theta$$

$$= \int_0^{2\pi} \cos(\theta - \beta) \cos(\theta - \gamma)\, d\theta$$

$$\left.\begin{aligned} &= \cos\beta \cos\gamma \int_0^{2\pi} \cos^2\theta\, d\theta + \sin\beta \sin\gamma \int_0^{2\pi} \sin^2\theta\, d\theta \\ &= \cos\beta \cos\gamma\, [\pi] + \sin\beta \sin\gamma\, [\pi] \\ &= \pi(\cos\beta\cos\gamma + \sin\beta\sin\gamma) \end{aligned}\right\} \text{as found earlier}$$

but in this case $\beta \neq \gamma$ so that $(\cos\beta\cos\gamma + \sin\beta\sin\gamma) \neq 1$, whereas in the cases considered earlier $\beta = \gamma$ and $\cos\beta\cos\gamma + \sin\beta\sin\gamma = \cos^2\beta + \sin^2\beta = 1$

Therefore the u.m.p. due to the third term will be

$$= \frac{lr}{2\mu_0} \frac{\epsilon}{2} B_p^2 \int_0^{2\pi} \cos\{\theta - (2\omega - \omega_\epsilon)t - (2\phi - \phi_\epsilon)\} \cos(\theta - \omega_\epsilon t - \phi_\epsilon)\, d\theta$$

$$= \frac{lr}{2\mu_0} \frac{\epsilon}{2} B_p^2 \pi (\cos\beta\cos\gamma + \sin\beta\sin\gamma)$$

$$= \frac{\pi r l B_p^2}{4\mu_0} \epsilon (\cos\beta\cos\gamma + \sin\beta\sin\gamma)$$

Now $\gamma = \omega_\epsilon t + \phi_\epsilon$ and $\beta = \{(2\omega - \omega_\epsilon)t + (2\phi - \phi_\epsilon)\} = \{(2\omega + 2\phi) - \gamma\}$

Therefore the u.m.p. due to this third term will not be a steady u.m.p. (which was the case for the first two u.m.p. contributing terms). The u.m.p. due to this third term will be a complex vibratory type of u.m.p. dependent on twice supply frequency and on the angular velocity of the rotating eccentricity. For static eccentricity the vibratory character of the u.m.p. due to this third term will depend on twice supply frequency alone.

It is interesting to compare this analysis with that of [31] MEILER, SPERLING and TIKVICKI. They sought to demonstrate analytically the existence of a type of vibratory oscillation of double supply frequency associated with static eccentricity. To achieve this they introduced a small backward rotating component of the mmf into the analysis. On this basis they produced, for a 2-pole machine, an expression for a type of u.m.p. derived from a component having a frequency of double the supply frequency. This was in addition to the usual component of u.m.p. This has been mentioned earlier in the section "Mmf wave". However, their final expression for this u.m.p. is not a vibrating force but a steady force, so they did not prove what they set out to demonstrate. It is, in fact, possible to show than an unbalanced vibrating force will result from the assumptions made by these authors but, in order to do so, it is necessary to produce an analysis along the lines adopted by the writer.

Furthermore, the writer has based his own work on simpler assumptions than those made by these authors. In particular he has not introduced a backward rotating component of mmf. On the basis of these simpler assumptions the writer has succeeded in deriving an expression for the type of complex vibratory u.m.p. that these authors would appear to have sought. This u.m.p., derived from the third u.m.p. contributing term of $B^2_{\theta,t}$, does not seem to have been described in the literature.

[29] That homopolar flux in a 2-pole machine may not be negligible is shown in a case reported by KOVACS (1977) in which 2-pole induction motor vibrations were attributed to homopolar alternating fluxes.

Also interesting was the observation of strong unbalanced magnetic pull in an induction motor in which homopolar flux was deliberately produced. The airgap of the motor was nominally concentric. The purpose of this experimental work was to study the effect of slotting on the stray losses. This work was recently done by J. SCHULTZ, a postgraduate student working under Professor N.C. ENSLIN at the University of Cape Town. [40][41][42][43]

2.18 Conclusions Regarding the Nature of Unbalanced Magnetic Pull:

As already shown, there are only three terms of $B^2_{\theta,t}$ which contribute to the u.m.p.

The first term is produced by the product $2B_p B_{p+1}$

[refer calculation $\overline{II}$. The term concerned is

$$\frac{\epsilon}{2} B_p^2 \cos(\theta - \omega_\epsilon t - \phi_\epsilon)]$$

The numbers of the pole-pairs of the harmonic fields differ by 1, that is, (p + 1) - p = 1.

The second term is produced by the product $2B_p B_{p-1}$ [refer calculation III. The term concerned is

$$\frac{\epsilon}{2} B_p^2 \cos(\theta - \omega_\epsilon t - \phi_\epsilon)]$$

The numbers of the pole-pairs of the harmonic fields differ by 1, that is p - (p-1) = 1.

The third term is also produced by the product $2B_p B_{p-1}$ [refer calculation III. The term concerned is

$$\frac{\epsilon}{2} B_p^2 \cos\{(2p-1)\theta - (2\omega - \omega_\epsilon) - (2\phi - \phi_\epsilon)\}]$$

The numbers of the pole-pairs of the harmonic fields differ by 1, that is, p - (p-1) = 1.

For the reasons given above it can now be stated that unbalanced magnetic pull between the stator and the rotor results from the products of harmonic magnetic fields having numbers of pole-pairs differing by unity.

Because the unbalanced magnetic pull results from the product of two such fields it is a modulation effect, that is, the two fields modulate each other. This is in accordance with the general equation for sinusoidal modulation,

$$\begin{aligned} y &= (y_1 + y_2 \cos \omega_2 t)(y_3 \cos \omega_3 t) \\ &= (y_1 y_3 + y_2 y_3 \cos \omega_2 t) \cos \omega_3 t \\ &= (a + b \cos \omega_2 t) \cos \omega_3 t \\ &= a \cos \omega_3 t + b \cos \omega_2 t \cos \omega_3 t \\ &= a \cos \omega_3 t + \frac{b}{2} \cos(\omega_2 + \omega_3) t + \frac{b}{2} \cos(\omega_2 - \omega_3) t \end{aligned}$$

It is necessary to state that the unbalanced magnetic pull does not result from interference or beating between the different harmonic magnetic fields. Such interference or beating would occur only as a result of the sum of two sinusoidal waves, say y_1 and y_2, where

$$y_1 = q \cos \omega_3 t$$

$$y_2 = q \cos(\omega_3 + \Delta\omega) t = q \cos \omega_4 t$$

giving

$$y_1 + y_2 = 2q \left(\frac{\omega_3 - \omega_4}{2}\right) t \cdot \cos\left(\frac{\omega_3 + \omega_4}{2}\right) t$$

The expressions given above for modulation and for beating are those usually stated on a time variable basis (t variable). The same considerations apply for a space variable basis (θ variable).

Modulation is different from beating even although angular differences are produced in both cases.

[49, p 79]

In considering the effects of eccentricity VON KAEHNE states that the harmonic fields interact giving force waves of various orders.

$$r = \Delta p = p_1 \pm p_2 = p(\nu \pm \mu)$$

He does not state how the fields interact, whether by modulation or beating.

In considering u.m.p. depending on slot combinations he states that certain investigators considered the combinations of two harmonic fields with νp and μp pole-pairs, p being the pole-pair number of the machine. In their mathematical treatment the investigators derived interference equations leading to a beating wave which causes u.m.p. The writer presumes that this is a beating wave of force and he must point out that, in the case of airgap eccentricity, such an u.m.p. producing beating wave of force does not appear to be present, nor does it result from the interaction of the harmonic fields. The u.m.p. resulting from airgap eccentricity is not produced by such a beating wave of force, but is, in fact, produced because two harmonic magnetic fields having numbers of pole-pairs differing by unity modulate each other.

The subject of u.m.p. depending on slot combinations requires further examination but this falls outside the scope of this thesis.

2.19 Forces other than unbalanced magnetic pull:

In the last section the calculation of unbalanced magnetic pull was studied. This focussed attention on the terms which it was proved contributed to the u.m.p. Attention will now be given to all the forces present because of the possible influence of any or all of these forces on the reliability of the motor.

The detailed expression of $B^2_{\theta,t}$ earlier in this thesis provides an opportunity to examine the magnitude, frequency and number of poles in each force wave as well as to examine how modulations of vibration can arise.

The components of $B^2_{\theta,t}$ will now be discussed.

The components of $B^2_{\theta,t}$ are representative of the force waves because $F_{\theta,t} = \dfrac{B^2_{\theta,t}}{2\mu_0}$.

In this discussion each component will be written without stating the factor B^2_p, this being a common factor of all components.

Term I:

$$= \frac{1}{2} + \frac{1}{2} \cos(2p\theta - 2\omega t - 2\phi)$$

The amplitude of these components is greater than that of any other components.

Example : Each component contributing u.m.p. in II and III has an amplitude $\frac{\epsilon}{2}$, where $\epsilon < 1$

Therefore $\dfrac{\text{amplitude of each component in term I}}{\text{amplitude of each u.m.p. contributing component.}} = \dfrac{\frac{1}{2}}{\frac{\epsilon}{2}} = \dfrac{1}{\epsilon} > 1$

The first component in term I is $\frac{1}{2}$.

This is a constant component and is independent of θ. This represents a constant, non-varying radial attraction between the stator and the rotor. This force would be a radial force on the rotor, pulling its surface outwards in the same way as does centrifugal force, and therefore this force tends to burst the rotor. Also this force would be a radial force on the stator core pulling it inwards. The radial attraction between the stator and rotor would have the effect of constricting the boundary surfaces of the airgap, thereby reducing the size of the airgap. The presence of this force does not appear to have been reported in the literature.

The second component in term I is $\frac{1}{2} \cos(2p\theta - 2\omega t - 2\phi)$. At any given position on the periphery of the airgap, that is, when has a definite fixed value, the only variable in this component will be the time t . Therefore, this component will vary in value in accordance with the angular velocity $2\omega\ (=2\pi f)$. Therefore, this component represents a force of twice supply frequency. Vibration at twice supply frequency is well known and has been reported in the literature. This force wave has 2p pole-pairs. In a 2-pole motor this force wave will have 2 pole-pairs. The significance of this is that, as one takes measurements around the periphery of the airgap, there will be a time phase difference between two positions dependent on the 2p pole-pair space relationship.

The interaction of the component of value $\frac{1}{2}$ and the component of the value $\frac{1}{2} \cos(2p\theta - 2\omega t - 2\phi)$ is shown in Fig. 4. This indicates that the whole of term I, that is $\frac{1}{2} + \frac{1}{2} \cos(2p\theta - 2\omega t - 2\phi)$ represents a combination of a sinusoidal vibration and a steady force. What is present is not simply a sinusoidal vibration at twice supply frequency about a neutral unstressed condition of the stator frame and rotor. The sinusoidal vibration, in fact, takes place in the presence of a steady force of attraction between the stator and the rotor. The steady force of attraction between the stator and the rotor is represented by the force magnitude factor $\frac{1}{2}$. The sinusoidal vibration is represented by a force magnitude factor $\frac{1}{2}$. The peak force is represented by a force magnitude factor 1. The total force $\frac{1}{2} + \frac{1}{2} \cos(2p\theta - 2\omega t - 2\phi)$ never becomes negative at any instant.

Term I is independent of the eccentricity, therefore it will be present in all induction motors.

The remaining terms (II to VI) are dependent on ϵ or ϵ^2, so they will only be present if eccentricity is present.

The significance of the terms contributing to the u.m.p. has already been discussed. The remaining terms will not be discussed or analysed individually. Such a discussion would be along the lines of the analysis of Term I which has just been given above. However, the frequencies of all the components of the force waves will now be discussed. The force waves have frequencies represented by the following angular velocities if rotating eccentricity is present:-

TERM NO.	FIRST COMPONENT OF TERM	SECOND COMPONENT OF TERM
I	0	2ω
II	$2\omega + \omega_\epsilon$	ω_ϵ
III	$2\omega - \omega_\epsilon$	ω_ϵ
IV	0	$2(\omega + \omega_\epsilon)$
V	2ω	$2\omega_\epsilon$
VI	0	$2(\omega - \omega_\epsilon)$

In addition to steady forces of zero frequency vibrations of the following frequencies should be detectable as a result of the above force waves:-

2ω : Twice supply frequency (100 Hz in a 2-pole motor).

$2\omega + \omega_\epsilon$: Approximately 3 x supply frequency in a 2-pole motor (approximately 150 Hz in a 2-pole motor).
Approximately $2\omega + \frac{\omega}{2} = \frac{5}{2}\omega$ in a 4-pole motor.
Approximately $2\omega + \frac{\omega}{p} = \omega(2 + \frac{1}{p})$ in a p pole-pair motor.

ω_ϵ : Rotational frequency.

$2\omega - \omega_\epsilon$: Approximately supply frequency in a 2-pole motor (approximately 50 Hz in a 2-pole motor).
Approximately $2\omega - \frac{\omega}{2} = \frac{3}{2}\omega$ in a 4-pole motor.
Approximately $2\omega - \frac{\omega}{p} = \omega(2 - \frac{1}{p})$ in a p pole-pair motor.

$2(\omega + \omega_\epsilon)$: Approximately 4 x supply frequency in a 2-pole motor (approximately 200 Hz in a 2-pole motor).
Approximately $2(\omega + \frac{\omega}{2}) = 2 \cdot \frac{3}{2}\omega = 3\omega$ in a 4 pole motor.
Approximately $2(\omega + \frac{\omega}{p}) = 2\omega(1 + \frac{1}{p})$ in a p pole-pair motor.

$2\omega_\epsilon$: Twice rotational frequency.

$2(\omega - \omega_\epsilon)$: 2 x slip frequency in a 2-pole motor.
Approximately $2(\omega - \frac{\omega}{2}) = \omega$ in a 4-pole motor.
Approximately $2(\omega - \frac{\omega}{p}) = 2\omega(1 - \frac{1}{p})$ in a p pole-pair motor.

The force waves have frequencies represented by the following angular velocities if only static eccentricity is present:-

TERM NO.	FIRST COMPONENT OF TERM	SECOND COMPONENT OF TERM
I	0	2ω
II	2ω	0
III	2ω	0
IV	0	2ω
V	2ω	0
VI	0	2ω

In addition to the steady forces of zero frequency the only vibrations indicated are those at 2ω, that is, twice supply frequency.

The writer's experimental measurements may be referred to in respect of the frequency analysis of large 4-pole motors. The principal components of vibration on bearings were at rotational speed with a smaller component at twice rotational speed. In some cases a component of roughly half rotational speed was present. A component at twice supply frequency was detected on a motor frame but this component was not detectable on the bearings. RAI's experimental observations of vibratory forces has already been mentioned. He conducted extensive laboratory tests into the vibratory forces due to static and dynamic eccentricity on a 2-pole motor under various operating conditions and three types of rotor, namely, unslotted, wound and cage rotor. Vibratory forces due to dynamic eccentricity only were studied on a small 6-pole cage rotor motor. For measurements of vibration due to static eccentricity in a 2-pole motor he reported that 50, 100 and 200 Hz components of vibratory force were most prominent and were found to change with both eccentricity and voltage. The magnitude of the 150 Hz and 300 Hz components were relatively small and did not bear any simple relationships with eccentricity or voltage. The possibility of these components being associated with mechanical unbalance was suggested. [33] [33, p107]

The theory which the writer has presented above predicts for static eccentricity the presence of vibratory forces at twice supply frequency, that is, 100 Hz, but not the 50, 150, 200 and 300 Hz components measured by RAI. RAI suggested by way of brief physical explanations that vibratory forces at frequencies other than 100 Hz could occur due to resonance between the mechanical system and the electromagnetic driving forces.

For measurements of vibration due to dynamic eccentricity in 2-pole and 6-pole motors RAI reported that the spectrum of vibratory forces mainly consisted of a component at rotational speed i.e. 50 Hz in his 2-pole machine and 16, 67 Hz in his 6-pole machine. The magnitude of the component at rotational speed was the largest compared to the other components. Vibratory forces at twice rotational speed, line and twice line [33, pp108, 120] frequency also occurred but their magnitude was quite small. It is [33, p 45] interesting to note that RAI measured the vibratory forces by means of the same force tranducers as he used for measuring the radial forces between rotor and stator in order to find the unbalanced magnetic pull. There were two sets of transducers, one set at each bearing, between the bearing housing and the main frame. With this arrangement he would not have been able to measure "balanced" vibrations such as can be detected by measurements of vibration on the motor frame using a vibration meter or vibration analyser. RAI's [33, p 120] measurement of vibration due to dynamic eccentricity in the 2-pole motor was found to have five frequencies, namely 50, 100, 150, 200 and 300 Hz, as for static eccentricity. He reported a modulation at twice slip frequency on the main component. The magnitude of the main component at rotational speed was the largest as compared to the other components. [33, p 125] The vibration due to dynamic eccentricity in the 6-pole motor having a cage rotor comprized the main component at rotational speed, that is, 16,67 Hz and vibration at twice rotational speed, that is, at 33,3 Hz, also at 50, 100, 150 and 250 Hz and at twice slip frequency. The amplitude of these components was in all cases less than 1% of the main component at 16,67 Hz.

The theory which the writer has presented above for rotating eccentricity predicts the presence in a 2-pole motor of the components of vibration at frequencies corresponding to rotational speed, twice [33, p 120] rotational speed, 50, 100, 150 and 200 Hz as found by RAI, but not the 300 Hz component. Also, modulation of the main component at twice slip frequency is not predicted by this theory although reported by RAI. A component of vibration at twice slip frequency is predicted by this theory although not measured by RAI.

[33, p 125] For the 6-pole motor the theory predicts the components at rotational speed, 16, 67 Hz, and twice rotational speed, 33,3 Hz, and at 100 Hz but not at 50, 150, 250 Hz and twice slip frequency. The theory predicts additional frequencies not reported by RAI.

It would appear that, in the case of rotating eccentricity the magnitude of the main component of vibration at rotational speed is greater in relation to other components of vibration than the foregoing theory predicts. This suggests that the theory based on electromagnetic force waves alone is not a complete representation of the actual phenomena present in the motor.

[33, p 120] RAI reported "modulation" of vibration at two times slip frequency in his measurements on a 2-pole motor. The writer has reported his measurements of "modulation" at one and two times slip frequency on 4-pole motors. Similar reports by others in the "modulation" of vibration in 2-pole motors exist. [38]

However, the writer's theory does not predict any modulation of vibration. There is, however, a possibility of beating between the components of vibration which could produce beat frequencies which have previously been believed to be modulation. The theory will be examined in detail presently to ascertain if such beating at one or two times slip frequency is, in fact, predicted.

In the meanwhile it can be seen that there is a considerable measure of agreement between the foregoing theory and the results of experimental observations. On the other hand some observations have produced results not given by the theory and some theoretical results are not reported in observations. On this basis the theory has a factual grounding but requires further refinement.

The frequencies of the force waves $F_{\theta,t}$ (Terms I to VI), as represented by their angular velocities, will now be studied to see if beat frequencies of one or two times slip frequency result from interaction between the different force waves. This is done in the following tables showing the differences and sums of the frequencies of the electrical forces.

In these tables beat frequencies which are multiples of slip frequency are shown in darkly outlined boxes. The symbol 2 SF(2P) written in the box indicates that the beat frequency is two times slip frequency in a 2-pole motor. The symbol 2 SF(4P) written in the box indicates that the beat frequency is two times slip frequency in a 4-pole motor. Examination of the terms of $B^2_{\theta,t}$ in conjunction with these tables shows that in all cases where a beat frequency which is a multiple of slip frequency in a 2-pole motor is shown, this beat frequency arises from the interaction of two force waves, at least one of which disappears for no homopolar flux. RAI reports that in his test the "homopolar [33, p 66] flux was very small due to very high reluctance to its path - particularly as in this test an insulating ring was provided between the frame and the bearings". Yet he reported "modulation" of vibration at two times slip frequency. This could indicate that in a commercial motor, where no special precaution is adopted to prevent homopolar flux, the strength of the homopolar flux actually present could be high enough to produce a detectable beat frequency of twice slip frequency and for this homopolar flux to make a considerable contribution to the magnitude of the unbalanced magnetic pull. It is usually considered that in a 2-pole motor the u.m.p. is one half the value of the u.m.p. present in a motor having more than 2-poles, because it is assumed that no homopolar flux is present in the 2-pole motor. For the foregoing reason the u.m.p. actually present in a commercial 2-pole motor could be higher than what has previously been assumed.

.20 Mechanical Vibratory Forces:

The writer will now examine the interaction between mechanical vibrations and the electromagnetic force waves to see if beat frequencies of one or two times slip frequency result.

The vibrational forces due to mechanical out of balance may be represented by

$$m_1 \cos(-\omega_t t - \phi_{m1}) + m_2 \cos(-2\omega_t t - \phi_{m2})$$

The fundamental component of mechanical driving force which arises from a rotating assembly is caused by mechanical static or dynamic unbalance. It has a frequency numerically equal to the speed of rotation in revolutions per second. The amplitude of this mechanical driving force may be controlled by careful mechanical static and dynamic balancing of the rotating assembly. In addition to the fundamental component, harmonic driving forces having frequencies which are multiples of the speed of rotation in revolutions per second frequently occur. These are caused

ELECTRICAL FORCES.

DIFFERENCES OF FREQUENCIES (LEFT COLUMN MINUS TOP ROW).

	0	2ω	$2\omega+\omega_\epsilon$	ω_ϵ	$2\omega-\omega_\epsilon$	$2\omega+2\omega_\epsilon$	$2\omega_\epsilon$	$2\omega-2\omega_\epsilon$
0	0	-2ω	$-2\omega-\omega_\epsilon$	$-\omega_\epsilon$	$-2\omega+\omega_\epsilon$	$-2\omega-2\omega_\epsilon$	$-2\omega_\epsilon$	2SF(2P) $-2\omega+2\omega_\epsilon$
2ω	2ω	0	$-\omega_\epsilon$	$2\omega-\omega_\epsilon$	ω_ϵ	$-2\omega_\epsilon$	2SF(2P) $2\omega-2\omega_\epsilon$	$+2\omega_\epsilon$
$2\omega+\omega_\epsilon$	$2\omega+\omega_\epsilon$	ω_ϵ	0	2ω	$2\omega_\epsilon$	$-\omega_\epsilon$	$2\omega-\omega_\epsilon$	$-\omega_\epsilon$
ω_ϵ	ω_ϵ	$\omega_\epsilon-2\omega$	-2ω	0	2SF(2P) $2\omega_\epsilon-2\omega$	$-2\omega-\omega_\epsilon$	$-\omega_\epsilon$	$-2\omega-\omega_\epsilon$
$2\omega-\omega_\epsilon$	$2\omega-\omega_\epsilon$	$-\omega_\epsilon$	$-2\omega_\epsilon$	2SF(2P) $2\omega-2\omega_\epsilon$	0	$-3\omega_\epsilon$	$2\omega-3\omega_\epsilon$	$+\omega_\epsilon$
$2\omega+2\omega_\epsilon$	$2\omega+2\omega_\epsilon$	$2\omega_\epsilon$	ω_ϵ	$2\omega+\omega_\epsilon$	ω_ϵ	0	2ω	$4\omega_\epsilon$
$2\omega_\epsilon$	$2\omega_\epsilon$	2SF(2P) $2\omega_\epsilon-2\omega$	$\omega_\epsilon-2\omega$	ω_ϵ	$3\omega_\epsilon-2\omega$	-2ω	0	2SF(4P) $4\omega_\epsilon-2\omega$
$2\omega-2\omega_\epsilon$	2SF(2P) $2\omega-2\omega_\epsilon$	$2\omega_\epsilon$	$-3\omega_\epsilon$	$2\omega-3\omega_\epsilon$	$-\omega_\epsilon$	$-4\omega_\epsilon$	2SF(4P) $2\omega-4\omega_\epsilon$	0

ELECTRICAL FORCES.

SUMS OF FREQUENCIES (LEFT COLUMN PLUS TOP ROW).

	0	2ω	$2\omega+\omega_\epsilon$	ω_ϵ	$2\omega-\omega_\epsilon$	$2\omega+2\omega_\epsilon$	$2\omega_\epsilon$	$2\omega-2\omega_\epsilon$
0	0	2ω	$2\omega+\omega_\epsilon$	ω_ϵ	$2\omega-\omega_\epsilon$	$2\omega+2\omega_\epsilon$	$2\omega_\epsilon$	2 SF(2P) $2\omega-2\omega_\epsilon$
2ω	2ω	4ω	$4\omega+\omega_\epsilon$	$2\omega+\omega_\epsilon$	$4\omega-\omega_\epsilon$	$4\omega+2\omega_\epsilon$	$2\omega+2\omega_\epsilon$	$4\omega-2\omega_\epsilon$
$2\omega+\omega_\epsilon$	$2\omega+\omega_\epsilon$	$4\omega+\omega_\epsilon$	$4\omega+2\omega_\epsilon$	$2\omega+2\omega_\epsilon$	4ω	$4\omega+3\omega_\epsilon$	$2\omega+3\omega_\epsilon$	$4\omega-\omega_\epsilon$
ω_ϵ	ω_ϵ	$2\omega+\omega_\epsilon$	$2\omega+2\omega_\epsilon$	$2\omega_\epsilon$	2ω	$2\omega+3\omega_\epsilon$	$3\omega_\epsilon$	$2\omega-\omega_\epsilon$
$2\omega-\omega_\epsilon$	$2\omega-\omega_\epsilon$	$4\omega-\omega_\epsilon$	4ω	2ω	$4\omega-2\omega_\epsilon$	$4\omega+\omega_\epsilon$	$2\omega+\omega_\epsilon$	$4\omega-3\omega_\epsilon$
$2\omega+2\omega_\epsilon$	$2\omega+2\omega_\epsilon$	$4\omega+2\omega_\epsilon$	$4\omega+3\omega_\epsilon$	$2\omega+3\omega_\epsilon$	$4\omega+\omega_\epsilon$	$4\omega+4\omega_\epsilon$	$2\omega+4\omega_\epsilon$	4ω
$2\omega_\epsilon$	$2\omega_\epsilon$	$2\omega+2\omega_\epsilon$	$2\omega+3\omega_\epsilon$	$3\omega_\epsilon$	$2\omega+\omega_\epsilon$	$2\omega+4\omega_\epsilon$	$4\omega_\epsilon$	2ω
$2\omega-2\omega_\epsilon$	2SF(2P) $2\omega-2\omega_\epsilon$	$4\omega-2\omega_\epsilon$	$4\omega-\omega_\epsilon$	$2\omega-\omega_\epsilon$	$4\omega-\omega_\epsilon$	4ω	2ω	4SF(2P) $4\omega-4\omega_\epsilon$

by mechanical imperfections or dissymmetries in the construction of the machine, particularly when the rotor has portions of non-circular cross-sectional area caused by keyways in the shaft, and when the bearings are slightly elliptic.

The interaction between the mechanical vibrations and the electromagnetic force waves is tabulated. In the table the beat frequencies which are multiples of slip frequency are shown in darkly outlined boxes and the symbol nomenclature is the same as has been explained in the case of the earlier tables indicating the interaction of the electromagnetic force waves alone.

It can be seen that the interaction of the mechanical and electrical forces results in beat frequencies of twice slip frequency in 2-pole and 4-pole motors.

One of the beat frequencies equal to twice slip frequency in a 2-pole motor arises from the interaction of the two forces having frequencies ω_ϵ and $2\omega - \omega_\epsilon$. The latter is present in a component in Term III which disappears for negligible homopolar flux. Therefore this beat frequency can be expected to disappear for negligible homopolar flux.

The second beat frequency equal to twice slip frequency in a 2-pole motor arises from the interaction of the two forces having frequencies 2ω and $2\omega_\epsilon$. The electrical force having a frequency 2ω will be present in all motors. The mechanical force having a frequency $2\omega_\epsilon$ will only disappear if the rotor is symmetrically constructed and balanced.

There is a dilemma here in connection with RAI's measurements. He reported that the vibration due to dynamic eccentricity in a 2-pole motor displayed a "modulation" at twice slip frequency on the main component, this being the rotational speed component. As has already been explained, he claimed that in his test rig the homopolar flux was very small. On this basis the "modulation" at twice slip frequency could not have occurred [33, p=42-43] due to the interaction of the electrical forces alone. However, the dynamic eccentricity in his test rig was produced by fitting an eccentric collar to the shaft at each bearing. A separate pair of collars was used for each eccentricity and after fitting these collars the rotor was accurately balanced each time so that any measured force was taken to result from electromagnetic causes alone.

Therefore, having eliminated the homopolar flux as well as the mechanical unbalance RAI should not have detected any "modulation" at twice slip frequency, yet he reported this "modulation" to be present. The writer would suggest that the elimination of homopolar flux and mechanical vibration may not have been as effective as RAI imagined. Possibly due to disymmetry of the rotor a mechanical vibration at twice rotational speed was present to cause beating at twice slip frequency.

The tables showing the interaction of forces indicate that a beat frequency of twice slip frequency in a 4-pole motor can result from either one or both of the following causes:-

1. Beating of two electrical force waves having frequencies $2\omega - 2\omega_\epsilon$ and $2\omega_\epsilon$ respectively.

2. Beating of a mechanical force wave having a frequency $2\omega_\epsilon$ with an electrical force wave having a frequency $2\omega - 2\omega_\epsilon$

MECHANICAL AND ELECTRICAL FORCES.

DIFFERENCES AND SUMS OF FREQUENCIES.

	DIFFERENCES (LEFT COLUMN MINUS TOP ROW)		SUMS (LEFT COLUMN PLUS TOP ROW)	
MECHANICAL FORCES →	ω_E	$2\omega_E$	ω_E	$2\omega_E$
ELECTRICAL FORCES ↓				
0	$-\omega_E$	$-2\omega_E$	ω_E	$2\omega_E$
2ω	$2\omega-\omega_E$	2SF(2P) $\boxed{2\omega-2\omega_E}$	$2\omega+\omega_E$	$2\omega+2\omega_E$
$2\omega+\omega_E$	2ω	$2\omega-\omega_E$	$2\omega+2\omega_E$	$2\omega+3\omega_E$
ω_E	0	$-\omega_E$	$2\omega_E$	$3\omega_E$
$2\omega-\omega_E$	2SF(2P) $\boxed{2\omega-2\omega_E}$	$2\omega-3\omega_E$	2ω	$2\omega+\omega_E$
$2\omega+2\omega_E$	$2\omega+\omega_E$	2ω	$2\omega+3\omega_E$	$2\omega+4\omega_E$
$2\omega_E$	ω_E	0	$3\omega_E$	$4\omega_E$
$2\omega-2\omega_E$	$2\omega-3\omega_E$	2SF(4P) $\boxed{2\omega-4\omega_E}$	$2\omega-\omega_E$	2ω

This is most significant from the point of view of this thesis because it provides a theoretical explanation for the phenomenon of beating at twice slip frequency in 4-pole motors. The writer has frequently observed and recorded this phenomenon in his experimental work on large motors in industrial use and on the test bed. Insofar as the writer can determine he has been the first to report such observations and to provide this theoretical explanation.

2.21 Advanced analysis of forces:

In addition to the work presented in this thesis, the writer has investigated the application of the more complex expressions for the mmf waves and permeance given earlier in this thesis to the calculation of the force waves. The presentation of these calculations falls outside the scope of this thesis and would form the subject of a more advanced study.

2.22 Conclusions:

The writer has presented a study of the electromagnetic and mechanical radial loading forces present in induction motors. The subject has been dealt with broadly with concentration on the electromagnetic forces due to airgap eccentricity, particularly the unbalanced magnetic pull and vibratory forces.

Insofar as the writer can determine this study provides the most explicit description of the phenomena to date and the mathematical theory is the most advanced. However a great deal of work remains to be done inthis field.

5.3 LOAD CARRYING CAPACITY OF INDUCTION MOTOR COMPONENTS:

5.3.1 LOADING OF BALL AND/OR ROLLER BEARINGS IN INDUCTION MOTORS.

The writer has, over a number of years, made searching enquiries into the design practices of several motor manufacturers with respect to the methods used to determine the size of ball and/or roller bearings in induction motors. These investigations have revealed that all these motor manufacturers have chosen the size of motor bearings on the basis of the well-known life formula

$$L = \left(\frac{C}{P}\right)^p \qquad \text{[32, p75] (PALMGREN's Formula 31.03)}$$

where,

C is the basic dynamic load rating of the bearing i.e. its carrying capacity in force units for a nominal life of 1 million revolutions (1 Mr)

P is the bearing load (equivalent bearing load) in force units.

L is the bearing life in millions of revolutions when subjected to the bearing load, P force units.

p is an exponent, p = 3 for ball bearings.
p = 10/3 for roller bearings.

While there are variations in the actual methods of calculation, all motor manufacturers base their design procedure on the above principle.

The bearing load is generally taken to be

$$P = W + UMP$$

where,

W is the contribution of the weight of the complete rotor to the loading of the bearing concerned. This weight includes any external loads on the shaft such as the weight of the motor half-coupling.

UMP is the contribution of the "unbalanced magnetic pull" to the loading of the bearing concerned.

Whilst W is easy to calculate, there are big differences between the values of UMP used by the different manufacturers, arising from different methods of calculating UMP.

All manufacturers conceive the u.m.p. as a steady one-sided magnetic pull in the direction of the smallest airgap. On the other hand, the writer has shown experimentally and theoretically that significant vibratory forces may be present and may act on the bearings.

An SKF handbook [1] advises that allowance should be made in the design of bearings in electrical machines for forces additional to any steady forces and gives factors for possible out of balance forces that may act on the bearings.

Possibly surprisingly, the writer has never come across a case where such an additional factor for possible out of balance forces has been used by any manufacturer in his bearing calculations in any 6600 volt motor.

PALMGREN [32] shows that, in addition to the calculation of bearing life on the basis of the life formula, it is often necessary to take into account the absolute maximum load F_{max} and the static equivalent load P_0, particularly if shock loads occur in service. PALMGREN [32] discusses in some detail the stresses and deformations due to contact between elastic bodies such as occurs in rolling type bearings. He then goes on to discuss the permanent deformations that can occur. In particular he shows that "if a bearing is subjected to a load while it is not rotating, the life formula cannot be used because it gives $P=\infty$ for $L = 0$. Obviously there is a limit to the load that the bearing can carry. This limit, however, is not determined by the effect of fatigue but by the permanent deformations which may develop in the load carrying surfaces or by the danger of fracture in various parts of the bearing" [32,p84] From considerations of the permanent deformations that can occur arises the concept of <u>static capacity</u> which is understood as "the load which a bearing can be subjected to while stationary, without deformations developing which would be noticeable when the bearing rotates under a lesser load and normal requirements for smooth running. The pure radial load or pure thrust load respectively which corresponds to this static load-carrying capacity of the bearing is called the <u>basic static load rating</u> and is designated by C_0.... [32,p85]

[32,p107] The maximum load on a rotating bearing can be allowed to exceed the basic static load rating provided that this maximum load acts without interruption while the bearing rotates several times. The small permanent deformations which develop are by this means evenly distributed over the raceways and the running quality of the bearing remains satisfactory .... If, on the other hand, the maximum load is of extremely short duration, as in the case of a pronounced shock load .... unevenly distributed deformations may develop even if the bearing is rotating at an instant when the shock is applied. In such a case, it is necessary to use a bearing whose static load rating is higher than the maximum load F_{max}".

From the above PALMGREN states [32,p108] that "the static capacity is important also for rotating bearings and that the relation between the maximum load and the static load rating is dependent on the character of the load variation".

The safety factor s_0 according to the formula

$$C_0 = s_0 P_0 \quad \text{.....................} \quad (43.01)$$

can thus be allowed to assume different values in different cases depending on the effect which the load variations have on the distribution of the deformations.

As a rule, the following coefficients apply:

$s_0 \geqq 0.5$ for smooth shock-free operation,

$s_0 \geqq 1$ for ordinary service,

$s_0 \geqq 2$ for sudden shocks and for high requirements on the smooth running of the bearing.

It is not possible to establish an exact permissible minimum value of s_0 since the static loading is not a specific limit of strength. However, a bearing which is to function under actual rotation with a speed worth mentioning, ought never, even temporarily, to be more heavily loaded than twice the static load rating $(s_0 = 0.5)$, while a bearing to which one applies particularly high requirements on smooth and quiet running, should not be more heavily loaded than half the static capacity $(s_0 = 2)$, regardless of how short a life might otherwise be permitted.

By combining formula (43.01) with the formula (31.03) the shortest applicable life is obtained:

$$L_{min} = \left(s_0 \frac{C}{C_0} \frac{P_0}{P}\right)^p \text{ - - - - - - - - - - - - - - - - } (43.02)$$

This formula shows that a bearing cannot always be dimensioned, guided solely by the exponential mean value of the load and by the life which is considered satisfactory. There is a minimum life for which the bearing may be dimensioned and this life depends, according to equation (43.02), on the bearing type; that is, on the ratio $\frac{C}{C_0}$, which has a different magnitude for bearings of different types and dimensions. This life is also dependent on the operating conditions as expressed in the formula by the factor s_0 and by the ratio $\frac{P_0}{P}$ ".

[32]

No apology is offered for quoting so extensively from PALMGREN because no clearer exposition can be found of the points the writer wishes to present.

From an examination of the angular velocities corresponding to the frequencies of the radial vibratory forces present in induction motors having airgap eccentricity, the writer's analysis has shown that some of these forces have values 2ω, $2\omega + \omega_\epsilon$, ω_ϵ, $2\omega - \omega_\epsilon$, $2(\omega + \omega_\epsilon)$, $2\omega_\epsilon$.

These are all forces of frequency greater than or equal to rotational frequency ω_ϵ. Therefore, the bearing does not rotate several times while the maximum value of each of these forces is applied. Permanent deformations developed in the bearings are not evenly distributed over the raceways. The maximum loads can be regarded as of extremely short duration and of the same nature as shock loads, in fact, shock loads of a cyclic character.

While it may be argued that the electromagnetic vibratory forces are of a "balanced" nature the effects of these forces on the bearings in motors of different types of construction is as yet not determined.

The writer's experimental work has shown that the vibrations measured on bearings in large 4-pole induction motors are dependent on the electrical supply and are not purely mechanical in nature. It is possible that the effect of the motor construction, which is not the same in all radial directions, may result in an unbalanced response at the bearings to the so-called "balanced" vibratory forces. The effects of the "balanced" vibratory forces in producing nodal deformations of the bearings corresponding to the nodal deformations of the stator and rotor is also not determined. In 2-pole motors with significant homopolar flux the writer has shown that unbalanced magnetic pull can be of a vibratory nature having twice supply frequency for static eccentricity. It is also certain that the mechanical unbalance forces having frequencies ω_ϵ and $2\omega_\epsilon$ act on the bearings.

The specific relationship of the electromagnetic and mechanical vibratory forces present in induction motors having airgap eccentricity to the life of ball and/or roller bearings installed in such motors requires further investigation.

5.3.2 LOADING OF JOURNAL BEARINGS IN INDUCTION MOTORS.

As for ball and/or roller bearings, the writer has over a number of years made searching enquiries into the design practices of several motor manufacturers with respect to the methods used to determine the size of journal bearings in induction motors.

These investigations have shown that each motor manufacturer has his own empirical figures for permissible bearing pressures. The bearing pressure is taken as the load applied per unit projected bearing area. The permissible bearing pressure is dependent on the peripheral speed of the shaft journal and on the method of bearing lubrication and cooling.

The bearing load P is generally taken to be

$$P = W + UMP$$

where P and UMP have already been defined in the Section "Loading of ball and/or roller bearings in induction motors".

One motor manufacturer's method is simply to consider P = W. He does not calculate UMP but considers the loading as being due to the static weight alone. The permissible bearing pressure is lower than normal because the UMP is not involved. The value of the permissible bearing pressure is based on the manufacturer's "experience".

As has already been stated there are big differences between the values of u.m.p. used by different manufacturers, these differences arising from different methods of calculating u.m.p. Also all manufacturers conceive the u.m.p. as a steady one-sided magnetic pull in the direction of the smallest airgap, whereas the writer has shown experimentally and theoretically that significant vibratory forces may be present and may act on the bearings.

SHAWKI and FREEMAN (1955) [44] have studied journal bearing performance under sinusoidally alternating and fluctuating loads.

They tested the performance of a complete journal bearing under a vertical cyclic load of the form $P = P_0 + P_1 \sin(\omega_1 t) + P_n \sin(n\omega_1 t + \psi)$. The bearing was lubricated with pure mineral oil supplied under pressure, the oil temperature being thermostatically controlled. The results of the experimental study were presented for the case where $P_n = 0$ and for values of $\frac{P_1}{P_0}$ between infinity and zero inclusive. Their test results showed that the ratio of load application to that of journal rotation $\frac{\omega_1}{\omega}$ is a prime factor in the bearing behaviour. Values of this ratio slightly less than 0,5 gave rise to critical changes in the bearing performance of a character mainly dependent on the ratio P_1 to P_0

A sinusoidally fluctuating load was taken as a steady load component P_0 superimposed on the fundamental P_1 . This corresponds to the situation in an induction motor.

The critical changes referred to were increases of journal friction and maximum eccentricity ratio suggesting the possibility that film breakdown could occur at the critical value of the speed ratio for more severe conditions that those of the tests.

Further investigation is required to determine if more recent work has been done in this field and to determine the specific relationship of the electromagnetic and mechanical forces present in induction motors to the load bearing capacity of oil lubricated sleeve bearings installed in such motors.

5.3.3 LOADING OF STATOR ASSEMBLIES OF INDUCTION MOTORS.

Several studies have been made of the effects of loading forces on the stator assemblies of induction motors with resulting vibration and noise.

DEN HARTOG (1928) [14][15] studied vibration in a type of frame (Fig. 5(a)) [14] which could be regarded as a ring with rigid ends (Fig. 5(b)) [14] or with hinged ends (Fig. 5(c)) [14] or with flexible ends (Fig. 5(d)) [15] Formulae for the natural frequency of vibration were calculated for each of these cases. The formula representing Fig. 5(d) contained the results representing Fig. 5(b) and Fig. 5(c) as limiting cases.

ERDELYI and HORVAY (1957) [21] studied the vibration modes present in a type of stator assembly (Fig. 6) comprising two concentric shells (the stator core inside the stator frame) coupled by non-rigid ribs.

ELLISON AND YANG (1971) [17] studied the natural frequencies of a stator assembly (Fig. 7) having a thick core coupled solidly through ribs to an outer thin frame.

The writer has experienced considerable trouble with vibration resulting in damage in large induction motors in service. These motors have had stator frames with weak features such as those described above. In general the problem was in the weakness of the outer stator frame (thin outer shell, thin outer frame, non-rigid ribs, non-rigid feet). The stator core in each case was therefore insufficiently supported against the action of the radial forces in particular.

By contrast, excellent results (extremely low vibration, quiet operation, no damage) has been obtained with the type of construction shown in Fig. 8.

This comprises a thick core coupled solidly through rigid ribs to a rigid outer frame. The rigid outer frame comprizes a set of heavy transverse plates usually 5 or 6 in number, as follows:-

One transverse plate forming each endshield	=	2 plates.
One transverse plate at each end of the stator core	=	2 plates.
One or two transverse plates at or near the middle of the stator core	=	1 or 2 plates.
		5 or 6 plates.

These transverse plates are held together by heavy longitudinal side plates in a box-like construction, these side plates giving rigidity against vertical and longitudinal deflection of the stator frame.

The endshield plates are further reinforced by additional heavy plates to prevent these endshield plates from buckling. Buckling of the endshield is possible because the bearing is usually outside the endshield and the load on the bearing has a cantilever effect. The reinforcing plates on the endshield are welded at right angles to the endshield plate vertically from top to bottom and horizontally from side to side.

This type of construction logically gives the motor great radial, transverse and longitudinal rigidity. A radial ventilation system is necessary with this type of construction because the presence of the transverse plates prevents axial ventilation outside the stator core.

The abovementioned investigators did not include calculations of the stator frame deflection in their publications.

DEN HARTOG's book [16] (4 editions, 1934, 1940, 1947 and 1956 respectively) discusses various aspects of vibration in the frames of electrical machines but also does not include calculations of the stator frame deflection.

ROARK and YOUNG's book [35] (fifth edition, 1975) provides formulae for stress and strain in many different types of structure, including formulae for circular rings. These formulae are adaptable to pipes and cylinders. The stator core, particularly the stator yoke behind the teeth can be considered as such a cylinder. Likewise, the stator outer frame is often in the form of such a cylinder, forming an outer concentric shell. For loading on such a ring by two opposing loads, each W, at diametrically opposite points in a vertical line, the changes in the vertical and horizontal diameters due to this loading (that is, the deflections of the ring) are shown to be respectively of the form

$$D_H = k_H \frac{WR^3}{EI}$$

$$D_V = k_V \frac{WR^3}{EI}$$

where E = modulus of elasticity.
I = moment of inertia of ring cross-section.
R = radius of the ring.
k_H and k_V are constants depending on the hoop deformation factor and on the transverse (radial) shear deformation factor.

This type of loading occurs in induction motors due to the magnetic attraction between the surfaces of the rotor and stator across the airgap. The greatest attraction between these magnetic surfaces occurs at points where the airgap flux density is a maximum, that is, at the centre of each pole. The attractive force is present whatever the polarity of a particular pole. There are always pole centres located diametrically opposite to one another. Therefore, there is always a pair of opposing loads acting at diametrically opposite points. Depending on the number of poles in the fundamental field and on the number of poles in each harmonic field there will be several such pairs of opposing loads at diametrically opposite points. In fact, the situation is highly complex because probably an infinite series of harmonic fields is present.

It is necessary to consider a different view point in the calculation of stator frame deflection. This is shown in a calculation which was submitted to the writer by a particular motor manufacturer. It may be noted that this has been the only manufacturer to submit a calculation of stator frame deflection to the writer. Other manufacturers have the attitude or they state that the frame has great rigidity or is "infinitely" stiff and they consider only shaft or rotor deflections. Often it is patently clear that the stator frame has very much less than great rigidity. The calculation submitted by the abovementioned manufacturer was made on the following lines.

The flexural rigidity K of the stator frame is taken as

$$K = \frac{48\,EI}{L^3}$$

where E = modulus of elasticity.
I = moment of inertia of cross-section of stator frame.
L = length of stator frame.
(This is the length between outer surfaces of the endshields. These endshields are basically flat plates).

The deflection δ of the stator frame is shown as

$$\delta = \frac{M_s}{K}$$

where M = Weight of stator core and windings + u.m.p.

Hence δ is calculated in a particular case as

$$\delta = 2{,}02 \times 10^{-4}\ \mathrm{cm}$$
$$= 2{,}02 \times 10^{-3}\ \mathrm{mm}$$
$$= 2{,}02\ \mathrm{microns}$$

δ is, in fact, the deflection of the stator frame considered as a beam. One end of the "beam" is considered to be supported on the drive-end foot-mountings. The other end of the "beam" is considered to be supported on the non-drive-end foot-mountings. The deflection of the stator frame is therefore regarded in the same way as the deflection of the shaft supported at its bearings. The motors for which the manufacturer made this calculation were of the type of construction shown in Fig. 8. Four of these slipring motors, each rated 3000 kW, 8 pole, 6600 volts were commissioned and have given perfect service. They are particularly smooth running. On no-load their levels of vibration measured on the motor bearings were less than 5% of the limit laid down in BS 2613 : 1970. It may be noted that δ as calculated above applies only to vertical deflection. These particular stators have comparatively heavy stator cores and furthermore the stator core is supported by rigid ribs to an outer frame comprising 6 heavy transverse plates, as per the description given relative to Fig. 8. Although the above calculation is not applicable to transverse or radial deflection (other than vertical), these transverse plates provide the stator frame with great rigidity against the action of transverse or radial forces.

ALGER's book [2, pp 367-370] (1965 edition) considers the stator frame vibration. For this purpose a formula is derived for the deflection due to radial forces. These radial forces are considered to be caused by the "train of harmonic fields superposed on the fundamental flux wave giving rise to high frequency pulsations in the radial magnetic forces". These are considered to be "resolved into a series of sinusoidal force waves with different numbers of poles, revolving at different speeds, each force wave having twice as many poles as the magnetic field that produces it ...... If the force producing magnetic field has two poles, there will be two opposite centres of maximum magnetic pull at the poles and two intermediate points of zero force. The stator will, therefore, be pulled into an elliptical shape, the short axis of the ellipse coinciding with the pole axis and revolving synchronously with it. This will give a four-node vibration. Similarly, a 2P-pole magnetic field produces a vibration with 4P nodes". In his calculation Alger neglects the uniform magnetic pull due to the average displacement so the result may be considered applicable to a "normal" motor with a perfectly concentric rotor.

The stator is considered as a hoop of steel. The nodal vibration displacement curve will have a point of inflection at each node so there will be zero bending moment at these points. The deflection of the stator can therefore be approximated as the deflection of a beam freely supported at each end and carrying a sinusoidally distributed load. This "beam" is considered to have the same cross-section as the stator yoke, the stiffening due to the stator teeth and frame being neglected. The beam deflection formulae used by ALGER are similar in form to those of ROARK and YOUNG for circular rings, the difference being in the values of the constants.

ALGER [2,p19] derived the following expression for the attractive force between the magnetic surfaces enclosing the airgap:-

$$F = 1{,}387 \; B^2 \times 10^8 \text{ pounds per square inch.}$$

He stated the following to be typical values:-

F = 36,8 pounds per square inch for 0,8 webers per square metre. [2,p20]

F   22   pounds per square inch for 4×10^{-4} webers per square inch. [2,p367]

Substituting this expression for F in place of the load W in the beam formula he calculated the deflection as [2,p370]

$$d = \frac{0{,}108 \; B^2 D D_s}{P^4 h^3} \text{ inch}$$

where   h = radial depth of stator core behind the slots, in inches.

= mean diameter of the stator core, in inches

= D at gap + 2 (slot depth) + h

P = number of pole-pairs.

B = peak airgap flux density, webers per square inch.

Considering the actual deflection of a curved ring under sinusoidal radially applied force for the particular nodal deformations corresponding to the number of pole-pairs of the harmonic field concerned, the formula given above on the basis of simple beam theory is changed by replacing the coefficient 0,108

by   0,193 for P = 1

by   0,137 for P = 2

by   0,123 for P = 3

[2, p370] ALGER's equation gives "the single amplitude of (non-resonant) radial vibration of the stator surface due to the force wave of a 2P-pole magnetic field. With given values of B, h and D, the vibration amplitude varies inversely as the fourth power of the number of poles". Hence, in a given motor, the fields with the fewest poles will tend to produce the most vibration.

Further investigation is necessary to determine what the stator deflection would be if the unbalanced magnetic pull due to the eccentricity were also taken into account.

Whatever sound formulae are derived for the deflection of the stator frame the writer would consider that the value of deflection to be aimed for should be about 2 microns, as resulted from the above quoted manufacturer's calculation. The writer draws this conclusion from the satisfaction given by the motors concerned in actual service. Also, the writer's present standard is to try to achieve vibration levels on the bearings of 5% or less of the BS 2613 : 1970 [11] vibration limit. Certain manufacturers have accepted this as an aim and in fact have usually achieved significantly lower levels of vibration than the 5% of BS limit, but they will guarantee the achievement of vibration only as low as about 25% of BS limit. (Refer "Enquiry Documents for 6600 volt Induction Motors" forming a part of this thesis).

There are several complicating factors in comparing levels of vibration on bearings with the levels of vibration on stator frames. However, the writer would specify that the level of vibration on the stator frame should be significantly less than 5% of the BS limit. The BS limit is applicable to vibration on bearings only. BS 2613 : 1970 [11] and BS 4999 : Part 50 [12] : 1972 do not state limits for vibration on stator frames. The writer would further specify that the calculated stator deflection should correspond to the low vibration displacement figure specified above to be less than 5% of the BS limit.

It may be noted that the considerations which have been discussed in this section with regard to the rigidity of the stator can be applied to the rigidity of the rotor, particularly the stiffness of the stator core (or rotor core) considered as a ring under the action of diametrically opposing forces. The strength of the stator core yoke (or rotor core yoke) behind the teeth and slots is of fundamental importance apart from the strength of the stator outer frame (or rotor inner support, say the shaft spider). Cutting down on the amount of the magnetic material in the yoke of the stator (or of the rotor) will not only influence the magnetic circuit and its possible saturation but will also reduce the structural rigidity of the stator (or rotor).

The above discussion has been confined to the loading of external forces on the stator (or rotor) assembly. Consideration should also be given to internal structural stresses and to the method of fabrication.

The writer has found that certain large motors are fabricated by structural welding and then the structure (stator or rotor) is machined without stress relieving. These motors have, in the writer's experience, been particularly unsuccessful, subject to vibration, bearing failures and associated problems. This is hardly surprising if one considers the structural distortion that can occur over a period of time due to the gradual relief of internal stresses after machining has taken place. Such distortion can easily cause airgap eccentricity and, therefore, unbalanced magnetic pull. On the other hand, other motor manufacturers will not consider machining unless the fabrication is first stress relieved by heat treatment. Such stress relieving is done in accordance with the requirements of Lloyd's Register of Shipping whereby "stress relieving is to be done by heating the welded structure uniformly and slowly to a temperature between 580° C and 650° C holding that temperature for not less than one hour per 25 mm of maximum plate thickness and thereafter allowing the structure to cool slowly in the furnace". These motors have been successful, with low levels of vibration and completely reliable.

In some large motors extensive use is made of staggered intermittent welding in the structural fabrication. Often welding is done on only one side of a joint where such a joint may be very long, that is, almost as long as the motor. These motors have been particularly unsuccessful, subject to vibration, bearing and associated problems. On the other hand, other manufacturers will not consider the use of such welding procedures but will use only continuous welding on both sides of each joint. These motors have been successful with low levels of vibration and completely reliable.

It is noteworthy that these successful motors had other features which are considered favourable in terms of the criteria of this thesis. On the other hand, the motors on which stress relieving was not done and in which intermittent or one-sided welding was done also had other features which are considered to be adverse in terms of the criteria of this thesis. Therefore one cannot be certain that these fabrication procedures are contributory to the problems that were experienced with these motors. However, it may be significant that so many different weak features should occur together in certain motors. The manufacturers concerned appear to be almost oblivious to the problems that can arise from airgap eccentricity, particularly to the unbalanced magnetic pull and the radial electromagnetic vibratory forces.

5.4. ADDITIONAL THEORETICAL CONSIDERATIONS:

5.4.1 THE EFFECTS OF A SQUIRREL CAGE ROTOR ON U.M.P.

BRADFORD (1968) [5][6] and RAI (1973) [33,p76] showed that the u.m.p. present in a motor having a squirrel cage rotor is much less than the u.m.p. in the same motor with a slipring rotor. This reduction in u.m.p. results from equalising currents set up in the cage by the non-uniform distribution of airgap flux density due to an eccentric airgap. By Lenz's Law, the rotor currents flow in such a direction as to set up magnetic fields opposing the fields which induced these currents. Therefore, the rotor currents maintain a more uniform distribution of airgap flux density. Consequently, the u.m.p. is reduced.

The conductors of a slipring rotor, being series connected, all carry the same current so it is not possible for equalising currents to flow. Therefore the non-uniform space distribution of the airgap flux density cannot be changed. Therefore the flow of currents in the rotor of a slipring motor cannot reduce the u.m.p.

The difference in u.m.p. between squirrel cage and slipring motors was known to earlier workers as discussed in VON KAEHNE's review (1963).[49, pp 17-24]

The practical effect of this in the writer's experience has been that troubles which could be attributed to u.m.p. have almost always manifested themselves in slipring motors but only rarely in squirrel cage motors.

5.4.2 THE EFFECTS OF PARALLEL PATHS IN THE STATOR WINDING ON U.M.P.

RAI (1973) [33, p77] showed experimentally that parallel paths in the stator winding reduced the u.m.p. considerably. The reduction in u.m.p. by parallel paths in a squirrel cage moor was not as large as in a slipring motor.

VON KAEHNE (1963) [49, p19] shows that ROSENBERG in 1918 already discussed the influence of parallel paths and that work in this direction was continued by ROBINSON in 1943. [34]

Professor N.C. ENSLIN (1977) [19][20] experimented with parallel paths in the stator while investigating the value of u.m.p. He measured the difference in the currents of the two parallel circuits of one of the phases. This difference was produced by interconnecting the secondaries of two current transformers and bridging this circuit with an ammeter. The circulating component of current produced by the eccentricity was limited by the comparatively high short circuit loop impedance of the stator. An interesting aspect of this current is that its variation with voltage is very similar to that shown by the mean value of u.m.p.

In discussion with the writer Professor ENSLIN suggested that this method of measuring circulating current could provide a means of monitoring eccentricity and u.m.p. in an induction motor.

5.4.3 PULSATIONS OF VIBRATION AT SLIP FREQUENCY.

The writer's experimental work has shown that on any bearing of any of the 4-pole induction motors tested vibration at twice slip frequency may be present. At other positions on the same bearing pulsations of vibration at slip frequency itself may be present.

The writer's theory shows that pulsations of vibration at twice slip frequency will be present in 2-pole and 4-pole induction motors under certain conditions. In 4-pole induction motors in particular it is necessary that rotating eccentricity be present. The writer's theory does not explain why pulsations at slip frequency itself should be present in a 4-pole induction motor.

SUMMERS (1955) [38] discussed vibration in 2-pole induction motors related to slip frequency and he gave physical explanations for these phenomena. Besides twice slip frequency pulsations he also discussed pulsations which vary directly with slip or as a more complex function. He stated that certain combinations of mechanical and electromagnetic forces could cause pulsations which vary with slip. In particular he considered this to be caused by a rotor which was mechanically dynamically unbalanced and which was not perfectly centered in the stator (that is, stationary eccentricity, therefore stationary unbalanced magnetic pull was present). He considered the running frequency vibration caused by dynamic unbalance to be modulated by the unbalanced magnetic force.

If SUMMERS is correct and his reasoning is applicable to 4-pole motors then it could mean that the writer's experimental observations of pulsations at twice slip frequency as well as at slip frequency itself on one and the same bearing could mean that both rotating and stationary eccentricities were present in the motor. Of course, this is quite probable.

5.4.4 CRITICAL SPEED CURVES.

It is well known that, in any machine having a rotating shaft or rotor, the vibration increases to a maximum at a speed called the "critical speed" More than one critical speed may exist. The effect is a mechanical one. The subject has been deeply studied and an extensive literature on vibration and critical speeds in mechanical machines exists.[25] An electrical machine is also a mechanical machine and is subject to vibration of mechanical origins.

The occurrence of a critical speed may be explained as follows.[37, pp 218-219] Should the centre of a massive rotor be displaced from the axis of rotation a centrifugal unbalanced force will act. This will produce a shaft deflection which will further increase the unbalanced force. The deflection will be resisted by the elastic shaft, which will develop a counter force. It will be seen that the rotor is considered here as a structure having mass and elasticity. Such a structure will have a natural frequency of vibration. Therefore, at some critical speed of rotation corresponding to the natural frequency of vibration the deflection will theoretically become infinite. In practice the deflection will be limited by friction, flexure loss or viscous damping, but at this critical speed the deflection will tend to be large. The curve of vibration displacement vs. shaft speed is ofter called the "critical speed curve".

RAI (1973) stated:[33, p 141] "It is generally accepted that the effect of airgap eccentricity is to reduce the critical speed. Various investigators have tried to establish a simple relation but no confirmation has been established by experiment. This needs further study".

FREISE and JORDAN (1962)[23] proved analytically that the unbalanced magnetic pull due to airgap eccentricity could be considered as a spring with a negative spring constant and that this negative spring constant ensures that the critical speed is shifted downwards. The writer's experimental results reported in this thesis confirm this analytical finding.

Motor designers give attention to the critical speed of the motor with the aim of obtaining low vibration at the normal running speed of the motor. They believe this aim is achieved by:-

1. separating the critical speed from the normal running speed by at least a certain minimum percentage of the normal running speed.

2. the presence of a sharp peak of the critical speed curve at a critical speed, that is, assuming that the vibration falls away sharply to low values at speeds removed from the critical speed.

This raises the question of how sharp this peak really is. Vibration theory shows that the higher the damping factor the less sharp is the critical speed curve. Motor designers generally consider that the material in the structure of a motor does not provide much damping and therefore the peak is sharp. The damping factor of a motor structure is considered to be fixed if the motor construction is fixed. Control of the damping factor is related to the material and structure of the motor. This material and structure of the motor is selected and designed by requirements other than vibration damping.

The other question which is raised is the effectiveness of the sharp peak characteristic of the critical speed curve to give low values of vibration at motor speeds significantly removed from the critical speed. The low damping required to produce a sharp peak could well permit undesirably high values of vibration to be present at motor speeds significantly removed from the critical speed.

The writer's experimental observations reported in this thesis show that the critical speed curves of the motors tested are such that, although these motors show higher vibration at the critical speeds than at other speeds, the vibration is nevertheless high over a wide range of speeds so that the vibration at the normal running speed is high. Also the critical speed curve is seen to be unsymmetrical about any critical speed. The curve starts at the origin (zero vibration at zero speed). The vibration increases progressively as the speed increases with superimposed peaking at the critical speeds. The first critical speeds were below the normal running speeds of these motors.

On the other hand, the writer has made measurements, not reported in this thesis, on successful motors having heavy stiff stator assemblies (as per Fig. 8) and heavy stiff rotors. In such motors there is no critical speed below the normal running speed of the motor. The level of vibration is far lower at all speeds up to the running speed of the motor than is the case in the motors of lighter and more flexible construction on which the measurements reported in this thesis were made. Clearly, the stiffness of these motors is high enough to ensure that the first critical speed is well above the normal running speed of the motor. The writer has not been able to establish whether the low level of vibration at all speeds up to the normal running speed of such a motor is simply due to this stiffness alone or whether it can also be attributed to damping and mass. The writer has established from manufacturers' data that, in the case of two different designs of such successful motors, the values of the first critical speeds were respectively in excess of 300% and 500% of the normal running speeds. Typically it is considered that the first critical speed should be at least 10% or 20% higher than the normal running speed when the critical speed is higher than the normal running speed. This points to the great stiffness in the aforementioned designs in elevating the first critical speeds to very high values well separated from the normal running speeds of the motors.

For larger output high-speed motors the rotor construction necessarily becomes more flexible. This is because of the mechanical limitations due to centrifugal forces imposed on the diameters of large high-speed machines. As a result of the increased flexibility of the rotor the first critical speed is below the normal running speed of the motor. The level of vibration at all speeds up to the normal running speed of the motor and the sharpness of the critical speed curve become especially important matters in this case. It may be noted that the unsuccessful motors on which the experimental observations were made as reported on in this thesis do not fall into this category and could well have been designed with stiffer stators and rotors to lift the first critical speed from below the normal running speed to well above the normal running speed.

The writer's investigations have led him to the following conclusions.

There exists a type of motor which may be regarded as successful. As compared with less successful motors such motors would have the following properties:-

1. High mechanical and structural reliability.

2. A comparatively low level of vibration throughout the normal speed range.

3. The stator construction is always extremely rigid.

4. The rotor construction is always extremely rigid except in larger high-speed motors where the rotor construction necessarily becomes more flexible.

In addition it is possible that the successful motors have a comparatively low level of vibration above the normal speed range and at each critical speed but this has not been proved.

With respect to the extensive literature on vibration, the wide variation in the vibration characteristics of motors points to a need for further investigation to show how a lower level of vibration is really attained in motor design by consideration of stiffness, damping, mass and the effect of electrical forces including unbalanced magnetic pull.

6. UNBALANCED MAGNETIC PULL IN THE DESIGN OF INDUCTION MOTORS.

The writer has, over a number of years, investigated the design practices of several motor manufacturers with respect to their methods of calculating the unbalanced magnetic pull and the application of the value of u.m.p. so calculated in the design of induction motors.

This investigation has shown that the assumptions made, the methods of calculating u.m.p. and the values of u.m.p. stated to be present vary enormously between different motor manufacturers. Such a wide variation in the stated values of u.m.p. would be found in tenders submitted in response to any one enquiry in which definite parameters of motor output, duty, supply and other conditions were given, as per the enquiry documents drawn up by the writer. These enquiry documents form a part of this thesis.

The main differences in the assumptions made by different motor manufacturers and the methods of calculating u.m.p. have been found to be:-

1. The theoretical bases differ between different manufacturers. VON KAEHNE[49], for example, has reviewed the methods of different investigators. He concludes that [49, p 21] formulae derived by these investigators all give more or less the same value of u.m.p. if one considers only eccentricity up to 20% and neglects all other influences like saturation, rotor currents, starting effects or parallel windings. If, however, these influences are considered, widely differing values are obtained. These differing theories are reflected in the different design practices.

   It may be noted that as regards the theoretical bases, all manufacturers in the writer's experience are influenced in their calculation of the u.m.p. by the "conventional approach" and they do not consider the "rotating wave approach". (Refer section 5 Theoretical Analysis).

2. Different values of "manufacturing tolerances" are assumed by different manufacturers, ranging from 5% eccentricity to 25% eccentricity. These values are the values of the "initial eccentricity", that is, the eccentricity before any power is applied to the motor. When the motor is connected to the supply, deflection of the motor assembly (mainly the shaft, rotor and stator) takes place due to the action of the u.m.p. This increases the eccentricity to a value which may be called the "final eccentricity" or "running eccentricity".

3. Different assumptions are made as to the law by which u.m.p. increases with eccentricity. This is influenced by the theory upon which the manufacturer bases his calculations. (Refer 1 above). Some manufacturers assume that the u.m.p. increases linearly with eccentricity. Others assume that the u.m.p. increases more than linearly with eccentricity. (Refer Fig. 9).

4. There are big differences regarding the assumed stiffness of the motor assembly, comprizing the shaft, rotor and stator. Stator frame stiffnesses are almost always taken to be infinite, whereas some frames are clearly very flexible. The stiffness of the motor assembly affects the deflection due to the forces and so influences the value of the "final eccentricity".

   It may be noted that structural differences vary widely between motors of different manufacture, for example, in the shaft diameter, the distance between bearing centres, the frame stiffness between motors of different design for a given output.

5. The interaction of the law of increase of u.m.p. with eccentricity (3 above), the stiffness of the motor assembly (4 above), and the initial eccentricity (2 above) result in the "final eccentricity" and hence result in the value of the u.m.p. corresponding to the final eccentricity. Some manufacturers represents this situation graphically along the lines indicated in Fig. 9. Others make a simple calculation of the u.m.p. directly from a formula on the basis of empirical assumptions.

6. There are particularly big differences in practice between different manufacturers as regards the degree of magnetic saturation of the motor. It is well-known that as the flux density B increases from zero the u.m.p. increases according to the B^2 law. (Fig.10). As

   B increases still more saturation begins to take effect so that the value of the u.m.p. falls below the value it would have attained if the B law had been followed. As B increases still more the u.m.p. reaches a maximum

   value. With still further increases in B the value of the u.m.p. decreases. Some manufacturers make use of this saturation effect by designing the flux density at rated voltage to be well into the saturated region. They therefore calculate unusually low values of u.m.p. The type of motor in which this approach is applied is generally light and flexible, for example, lighter bearing, lighter weight of rotor, lighter weight of stator, lower D^2L. This type of motor has, in the experience

   of the writer, been particularly unsuccessful in its reliability. Such motors have been the subject of unpublished investigations by the writer.

   This lightness and flexibility of construction is to an extent the result of the calculation of a lower value of u.m.p. This applies particularly to the design of shaft and bearings. To some extent this type of construction appears to be simply co-incidental with the above-described design approach (operation deep into the saturated region, lower calculated value of u.m.p.) and not the result of it. This applies particularly to the design of stator frame. The designers of these motors seem to give little attention to the presence of the radial forces and to the necessity of having an extremely rigid stator frame able to withstand these forces. The light and flexible stator frames may simply be the result of economic pressures and of design policies untempered by a conciousness of the vital necessity for great rigidity.

It is not certain whether the poor reliability of these motors can be attributed to lack of understanding of the nature and effects of the forces in the highly saturated region or whether the poor reliability is simply a result of the lightness of construction and lack of rigidity. The forces present in the motor when operating under highly saturated conditions require further investigation, particularly the vibratory forces and the effects of the saturation harmonic fields.

Certainly the most reliable motors do not work into the saturated region to any significant degree and are of heavier stiffer construction for a given output.

7. Some manufacturers use different values of u.m.p. for different aspects of the design. A higher value of u.m.p. corresponding to the peak of the u.m.p. vs B curve is used to calculate the shaft deflection because this is considered a necessary check against the possibility of rubbing taking place between the rotor and the stator. A lower value of u.m.p. corresponding to the u.m.p. at rated voltage is used to calculate the loadings on the bearings, this being considered an adequate assessment of the average loading which can affect bearing life. On the other hand, most manufacturers use a single value of u.m.p. for all calculations, that is, for shaft deflection and for bearing loading.

CONCLUSION:

Much further work is required in order to reach the position in which it would be possible to standardize assumptions and the methods of calculation of unbalanced magnetic pull. A similar effort is required to standardize allowances for the effects of balanced and possible unbalanced vibratory forces.

Until such standardization is achieved one cannot consider tenders for motors to be on a truly comparable basis.

The writer plans to continue working in this direction.

7. ENQUIRY DOCUMENTS, SPECIFICATIONS AND ASSOCIATED DOCUMENTS.

The writer has been led to the development and writing of documents aimed to protect the purchaser against the supply of unreliable induction motors.

The development of these documents become necessary as a result of an inordinate number of premature motor failures which were experienced in the writer's organization as well as the fact that none of the existing standard specifications offered any significant protection to the purchaser against the supply of unreliable motors.

The development of these documents went hand in hand with the writer's investigations into the adverse effects of unbalanced magnetic pull and associated phenomena.

The main documents which the writer developed and wrote are entitled "Enquiry Documents for 6600 volt Induction Motors". These Enquiry Documents comprize five separate but associated documents, as follows:-

1. Introduction to Documents.
2. Form of Enquiry.
3. Schedule of Conditions.
4. Specification.
5. Form of Tender.

In addition, the writer developed and wrote an associated document entitled "Schedule of Information Required from Supplier of Machinery to be Driven".

These documents are included as a part of this thesis.

During the course of their development these documents underwent several revisions over a period of years. The revision of "Enquiry Documents for 6600 volt Induction Motors" included in this thesis is the latest revision completed in the first quarter of 1977. The revision of "Schedule of Information Required from Supplier of Machinery to be Driven" is the latest revision completed in December, 1976.

These documents are due for further revision because the writer visualises several specific aspects which could be improved. As with all specifications and legal documents, amendment will always be necessary in the future as circumstances demand.

Nevertheless, the writer has been informed independently by a number of overseas motor manufacturers, who receive enquiries on a world-wide basis, that these documents are the most comprehensive and advanced documents they have received and that no other documents approach these documents in scope and content.

The response of motor manufacturers to these documents has been most encouraging. The motor manufacturer is expected to do much more work than normal in order to submit a tender in response to these documents. For this reason, some manufacturers have not submitted tenders in response to certain enquiries. This has usually been when the manufacturer concerned considered he could not meet the delivery date required or that he could not offer a competitive price for the size of motor concerned. However, regardless of this aspect there has been an increasing display of interest on the part of almost all local and overseas motor manufacturers who operate in the South African market to submit tenders corresponding as closely as possible to the requirements of these documents. Some manufacturers have, in fact, responded by adjusting their designs and their manufacturing methods to comply with these requirements. Other manufacturers who have not yet been favoured with an order on the basis of these documents have indicated that they would be willing to adjust their designs and their manufacturing methods to comply with these requirements.

The success of these documents has been dramatically demonstrated by the outstanding performance and perfect reliability demonstrated in service of all of the motors which were enquired for, ordered, manufactured, supplied, installed and commissioned in terms of these documents. These motors have been enthusiastically acclaimed by the staffs of the companies for whom they were purchased.

Apart from various motor manufacturers, local and overseas, to whom these documents have been issued, the documents have become known locally amongst motor users, and the writer has responded to requests for copies of these documents from a number of leading South African organizations concerned with the purchasing and specification of induction motors. These organizations aim to adapt these documents to suit their own requirements. At least two organizations have already included certain elements of these documents in their enquiries for induction motors. It is likely that several versions or adaptations of the writer's documents will be used increasingly by responsible purchasing organisations in South Africa.

These documents include all the principal factors involved in the selection of 6600 volt induction motors and therefore go well beyond the scope of this thesis. The reasons for the inclusion of these documents in their entirety in this thesis is that their development was associated with the writer's study of unbalanced magnetic pull and associated phenomena. The selection of only those portions of these documents relevant to this thesis would be impractical. It is furthermore considered that the inclusion of these documents in their complete form in this thesis would enhance the value of the thesis as a work of reference.

7.1 TENDER ANALYSIS.

The writer has devoted a great amount of time over a period of several years to the analysis of tenders submitted in response to enquiries for induction motors. This work has gone hand in hand with the development of enquiry documents, specifications and associated documents as well as with the study of the adverse effects of unbalanced magnetic pull and associated phenomena.

As progressively more details were called for, big differences in significant features of the motor designs were surprisingly revealed between the offers of different motor manufacturers. These big differences in design were found in tenders submitted in response to any one enquiry in which definite parameters of motor output, duty, supply and other conditions were given. The significant design features involved were items such as size of bearings, weight of rotor, weight of stator, D^2L, and $\frac{D}{L}$ ratio. To give an idea of how great the design differences can be, in a certain enquiry one manufacturer submitted an offer having a total motor weight only about 45% of that submitted by another manufacturer.

It was also found that there was no relationship between the prices quoted for motors and the technical merits of the motors. The writer has had many detailed discussions with several motor designers from local and overseas motor manufacturers. These designers were found to be always under the impression that the technical differences between motors made by different manufacturers were comparatively slight and that if one motor were marginally better than another motor in technical features it was bound to cost more. The writer must state that the impressions of these designers are completely incorrect. Such a line of thought may correctly apply within the organization of a particular manufacturer. It certainly does not apply when looking at the results of a detailed tender analysis. The motor which is technically the best may be the most expensive motor. Not infrequently the motor which is technically the best may be the cheapest motor.

By purchasing an unreliable motor the costs that would be incurred in addition to the initial capital cost of the motor would comprize direct repair and/or replacement costs as well as the costs of the consequential losses, say in production. These additional costs often amount to several times the initial capital cost of the motor. For this reason it is necessary when adjudicating tenders to take into account not only the initial capital cost of the motor, the delivery date, and the commercial terms and conditions but also the technical details.

Advantages of making a detailed technical tender analysis:

1. One can avoid the trap of automatically accepting the lowest tender. The technical analysis may show that the motor concerned may be technically inferior.

2. One may justify spending marginally more than the price of the lowest tender for a motor having vastly superior technical features.

3. One may be able to spot technical weaknesses in the motor of one's choice, even if this be the motor having the best technical features. By negotiation these technical weaknesses can be rectified before the adjudication is completed or, less desirably, after an order has been placed.

4. One is able to learn an enormous amount from the tenders submitted. In this way the enquiry documents become an information gathering instrument. This, in particular, has been the writer's approach and he has used this information to make improvements to successive revisions of the enquiry documents. The knowledge gained has been invaluable in the decision making process.

Disadvantages of making a detailed technical tender analysis:

1. Some work is involved. It is the writer's experience that usually those responsible for making tender analyses are unfortunately not willing, capable or interested enough to do this work and would prefer to purchase motors on a "black box" approach, even if the client subsequently suffers through the unreliability of these motors.

2. Some organizations are bound by rules to accept the lowest tender without regard to technical considerations. No really adequate specification exists so the enquiries are always on the basis of an inadequate specification. These purchasers may think that by adopting this procedure all tenders are submitted on an equal basis and that the motors offered are equivalent to each other. This is, in fact, very far from the truth. It is patently clear that such organizations lay themselves wide open to the purchase of unreliable motors and it is well-known that it is usually such unreliable motors which they purchase.

It is because of these disadvantages that the development of an easily usable standard specification that would protect the user against the purchase of an unreliable motor may be considered necessary. The writer's present work may be regarded as a step in this direction. He aims to proceed further but more research work is necessary before a standard specification which would be acceptable to both manufacturer and user can be developed.

On the other hand, the advantages of making a detailed technical analysis would be lost if such a standard specification were used.

This raises a problem of a more general nature. Much adjudication and choice of complex engineering equipment is done in an unprofessional manner by people not qualified to undertake this task. These people (and their employers) consider the equipment to be "simple".

Therefore, in order to enable a sound adjudication to be made on the basis of a thorough technical investigation the writer would consider it advisable to continue to use and develop the present detailed set of enquiry documents for professional use even after the simpler standard specification for more general use has been developed.

8. CONCLUSIONS.

Insofar as the writer can determine, much of the work reported in this thesis constitutes a new contribution to the experimental and theoretical knowledge of induction motors.

This applies particularly to the experimental studies of vibration, stator current and rotor current, to the theoretical analysis of the electromagnetic loading forces, including the unbalanced magnetic pull, and of the mechanical loading forces in induction motors, to the discussions on the loading of induction motor components and to the development of detailed enquiry documents and associated documents which include all the principal factors involved in the selection of 6600 volt induction motors.

Comparison between the results of the writer's experimental work and his theoretical work provides a correlation in respect of the pulsations of vibration which were found to be present in several large 4-pole 6600 volt horizontal spindle induction motors. These motors vibrated excessively and were subject to various premature failures involving damage to mechanical and structural components. It was found in these experimental observations that pulsations of vibration were present at slip frequency at some positions on such a motor and at twice slip frequency at other positions on the motor.

Insofar as the writer can determine he has been the first to report such observations on 4-pole motors. Similar pulsations in 2-pole motors have been reported in the literature, although not in the detail and not covering several aspects presented in this thesis. The literature has stated that the pulsations in 2-pole motors were a "modulation" of the vibration. Therefore, the writer was led to believe that the pulsations which he observed in 4-pole motors were "modulations" of vibration. He described them as such in his reports. However, the writer's mathematical development of the theory has shown that pulsations of vibration at twice slip frequency in both 2-pole and 4-pole motors arise from interactions between specific electromagnetic forces or between a specific electromagnetic force and a specific mechanical unbalance force resulting in a vibration beat frequency of twice slip frequency. In particular it is shown that a vibration beat frequency of twice slip frequency in a 4-pole motor can result from either one or both of the following causes:-

1. Beating of two electrical force waves having frequencies $2\omega-2\omega_\varepsilon$ and $2\omega_\varepsilon$ respectively.

2. Beating of a mechanical force wave having a frequency $2\omega_\varepsilon$ with an electrical force wave having a frequency $2\omega-2\omega_\varepsilon$

In both causes stated above ω_ε is present in the electrical force waves which contribute to the vibration beat frequency, indicating the presence of rotating eccentricity and therefore rotating unbalanced magnetic pull.

In addition the experimental work has shown that the vibration on the bearings drops to a low value almost immediately upon removing the electrical supply from the motor.

The writer has therefore proved both experimentally and theoretically that, in the motors tested, the electrical forces provide the major contribution to the vibration on the bearings and that a rotating unbalanced magnetic pull is present. The rotating unbalanced magnetic pull is thus seen to contribute to the destructive vibration and therefore to have an adverse influence on the reliability of these motors.

In the literature a physical explanation is given of pulsations of vibration at slip frequency in 2-pole motors. This is considered to be caused by a rotor which is mechanically dynamically unbalanced and which is not perfectly centred in the stator, that is, stationary eccentricity, therefore stationary unbalanced magnetic pull is present. If this reasoning is correct and is applicable to 4-pole motors then it could mean that the writer's experimental observations in several motors of pulsations of vibration at twice slip frequency as well as at slip frequency itself on one and the same bearing in any such motor could mean that both rotating and stationary eccentricities were present in the motors.

In addition the writer participated with others in measurements of the unbalanced magnetic pull and associated phenomena present in a large 4-pole motor. These measurements showed that the force (u.m.p.) vs. applied voltage (corresponding to magnetic flux density) was of the well known form shown in Fig. 10, but the rated voltage was found to be well beyond the critical voltage, that is, the machine operated deeply into the saturated region. It was found that as the airgap eccentricity increased the peak of the u.m.p. vs. applied voltage curve increased in value. There was a correlation between the abovementioned measurements and KVAR resultant unbalance measurements made at the same time. Measurements of u.m.p. on load revealed unexpected measurement difficulties but the results showed there was little if any significant increase in the value of the steady u.m.p. with load. These results were certainly not in agreement with experimental findings by RAI, who reported that the u.m.p. increased significantly with load. It is possible that RAI obtained this result because his motor did not operate as deeply into the saturated region as was the case in the larger motor tested. The tests on the larger motor showed that there was an increase in both the horizontal and vertical bearing vibration as the load increased.

The writer has conducted a careful examination of the basic assumptions upon which the theory is based, particularly the assumptions regarding the airgap size and the m.m.f. wave. This examination goes well beyond the simple assumptions upon which the mathematical theory is developed.

On the basis of the simple expression for airgap size, for rotating eccentricity the airgap variation with time will be the same in all radial directions. For an m.m.f. wave of constant magnitude this can be expected to produce forces of the same magnitude in all directions. For a motor constructed equally stiff in all directions the vibration produced would be the same in all radial directions. However, the writer's experimental observations showed that during normal operation of the motors tested the vibration was much greater in the horizontal than in the vertical direction. When carefully measured the line of maximum vibration was inclined at a slight angle to the horizontal. The greater magnitude of the horizontal vibration could possibly be explained by greater frame stiffness in the vertical than in the horizontal direction. However, there is a possibility that the variation in airgap length may have been substantially different to that postulated in the simple expression. For example, with significant bearing clearance and with the shaft resting on the bottom of the bearing more horizontal than vertical movement of the rotor may have been taking place. For this reason the writer has developed a more complex mathematical expression for the size of the airgap in such horizontal spindle motors which should give a truer description of what actually takes place.

Similarly, whilst the writer's mathematical development of the theory is based on a simple expression for the m.m.f. wave, consideration is given to more complex expressions which should be closer to reality.

On the basis of simple expressions for the airgap size and the m.m.f. wave this thesis provides detailed and rigid mathematical proofs for the expressions for the unbalanced magnetic pull present in different cases. An interesting feature is that this theory predicts that, in the presence of homopolar flux in a 2-pole motor, there will be present, in addition to the steady u.m.p., a complex vibratory type of u.m.p. dependent on twice supply frequency and on the angular velocity of the rotating eccentricity. For static eccentricity the vibratory character of this type of u.m.p. will depend on twice supply frequency alone.

The discussions in this thesis on the loading of bearings questions the current practice of designing bearings on the basis of a steady u.m.p. It is shown that allowances must be made for repeated shock loading, for maximum loadings and for perodically fluctuating loadings which are known to be present due to the combined action of electromagnetic forces and mechanical out of balance forces.

The discussions in this thesis on the loading of stator assemblies shows that such assemblies must have great rigidity to withstand normal electromagnetic forces as well as to withstand electromagnetic forces that can arise due to eccentricity. Also the fabricated assembly must be stress relieved to avoid distortion which can cause airgap eccentricity.

Much work remains to be done in this field both theoretically and experimentally. The experimental work reported in this thesis has produced a wealth of information which has been only partly explained. Nevertheless, insofar as can be determined, the theory provides the most explicit description of the phenomena to date and mathematically the theory is the most advanced.

Methods of monitoring u.m.p., eccentricity and related paramaters on operating motors have been proposed, as discussed in this thesis and elsewhere. These methods involve the use of special connections and equipment and such methods are not applicable to all motors. On the other hand vibration analysis is applicable to all motors. The present study of vibration has given a fresh and fascinating insight into the behaviour of induction motors. Further study of vibration in motors should lead to a deeper understanding of the processes and parasitic effects of electromagnetic and mechanical origin present in induction motors. This could provide new methods of monitoring u.m.p., eccentricity and related parameters.

The present work on large industrial motors has revealed aspects of motor behaviour not previously reported in studies on smaller motors, especially laboratory models. This is probably due to the greater flexibility of these large motors in relation to the forces present and possibly due to the clearances present in practical bearings. The need for further studies on large motors is indicated. This need is universal but is especially required in South Africa where the use of such large motors is common and of great economic importance. South African industry is predominantly heavy, essentially mining and mineral processing. It remains to be seen whether South African users and motor manufacturers will realise the necessity for such research and will give it the necessary support. Users have facilities for an aspect of such research on machines installed and already in use. Existing test bed facilities in the works of local manufacturers and repair firms are inadequate, limited in their capacities and capabilities and, in any case, heavily committed to productive work. This has been the writer's experienceand it is a conclusion and proposal of this thesis that the establishment of a heavy electrical machine test facility in a University is a goal to be aimed for.

Much research has aimed at the improvement of motor efficiency. This has been so successful that today only very marginal improvements in efficiency are possible. By contrast, enormous improvements in motor reliability are possible in many cases.

Today's demand for energy saving has resulted in a renewed interest in the improvement of motor efficiency. Some manufacturers have responded to this demand by the use of more material in motors. This is a reversal of the trend over many years to manufacture smaller motors for a given output. The latest trend to manufacture larger motors to achieve higher efficiency to an extent goes in the direction of satisfying the requirement for higher reliability.

For these reasons research aimed at the improvement of reliability must surely be economically worthwhile.

The success of the writer's approach, and particularly of the enquiry documents which have been developed, has been shown by the fact that all the motors which have been selected in terms of these documents and which have been put into service have demonstrated outstanding performance and perfect reliability. This dramatic reversal of the unfortunate trend which initiated the investigation is the most concrete support of the writer's thesis that the effects of the radial electromagnetic forces, including the unbalanced magnetic pull due to airgap eccentricity, must be carefully taken into account in the design and construction of induction motors in order to achieve high reliability.

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