

CRANFIELD UNIVERSITY



K B TAWI

INVESTIGATION OF BELT MISALIGNMENT EFFECTS ON
METAL PUSHING V-BELT
CONTINUOUSLY VARIABLE TRANSMISSION

SCHOOL OF MECHANICAL ENGINEERING

Ph.D. THESIS

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SCHOOL OF MECHANICAL ENGINEERING

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ABSTRACT

The existence of belt misalignment in the Van Doorne metal pushing V-belt continuously variable transmission (CVT) has always been recognised by researchers and the manufacturers themselves. However, its effect has always been assumed to be negligible and search through the literature reveals no work published so far to verify this assumption. Misalignment of the belt may increase losses and the need to improve efficiency is now of paramount importance.

This work is an initial effort by the author to address this area and study the geometrical behaviour of the belt misalignment theoretically. An equation to represent this geometrical behaviour has been developed and it is able to calculate the magnitude and position of the belt misalignment at different speed ratios and at different assembled zero belt misalignment speed ratios. The condition necessary to minimise the magnitude of the belt misalignment has been determined. Verification of the equation was obtained by using a solid modeller from Unigraphics Version 10.4.0 CAD package and by experimental results.

Proceeding from the above, another model to predict the belt misalignment effect on the transmission efficiency has been developed based on Coulomb friction theory. The model is able to predict the transmission efficiency trends reasonably well but not its absolute values. The results from the experimental work seems to agree only partially with the developed model. The main suspect of this disagreement would probably due to the inevitable belt slip phenomena of the belt and secondly, could be due to the vibration of the transmission which needs to be addressed in future experimental work. Nevertheless, this investigation has succeeded in showing that increasing belt misalignment does have a small effect on the transmission performance which is sometimes favourable and sometimes not.

The results of the experimental work on using optical instrumentation to measure belt misalignment, has illustrated the possibilities and potentials of the system to carry out the belt misalignment measurements to a reasonable accuracy.

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NOTATION AND ABBREVIATIONS

Symbol	Remarks	Units
α	angle as defined in Fig. 3.1	<i>rad</i>
α_s	pressure-viscosity coefficient between segment and pulley	m^2/N
β	angle of lap of belt around the pulley	<i>rad</i>
β_P, β_S	ideal angle of lap of the belt on the primary and secondary pulleys respectively	<i>rad</i>
δ	wrapping angle on smaller pulley; elemental symbol	<i>rad</i>
δd_p	segment strain length at the primary pulley	<i>mm</i>
$\delta d'_p$	segment strain length at the primary pulley during exit and entry from tangency point to take of point	<i>mm</i>
Δ	belt misalignment	<i>mm</i>
$\Delta_1, \Delta_2,$ Δ_{P1}, Δ_{P2}	belt misalignment as defined in Fig. 3.2	<i>mm</i>
Δ_{max}	maximum belt misalignment distance of the segment allowed by the proposed theoretical model	<i>mm</i>
ϵ_{segP}	segment strain at the primary pulley	-
μ	coefficient of friction between contact surfaces	-
μ_b	coefficient of friction between bands	-
η	viscosity of the oil; belt efficiency	$N.s/m^2; \%$
η_{os}	basic viscosity of lubricant between segment and pulley	$N.s/m^2$
η_s	viscosity of oil film between segment and pulley	$N.s/m^2$
ϕ	friction angle ($\phi = \tan^{-1} \mu$)	<i>rad</i>
φ	symbol for angle	<i>rad</i>
φ_{aP}, φ_e	angles as defined in Fig. 5.12 and 5.14 respectively	<i>rad</i>
φ'_e	angle as defined in Fig. 5.17	<i>rad</i>
σ_{segP}	segment stress at the primary pulley	N/m^2

Σ	summation symbol	-
τ	viscous shear stress	N/m^2
θ	half groove angle at V-belt pulley	rad
ϑ	symbol for angle	rad
ϑ_{mP}	angle of optical measuring system	deg
a	segment thickness	mm
δ	additional belt misalignment spacer	mm
b	width of segment contact area	mm
c	height of segment contact area	mm
d	unrestrained segment width at centre of contact area	mm
d_b	band width	mm
d_m	unrestrained segment width at rocking edge	mm
d'_p, d'_s	compressed segment width at the primary and secondary pulleys	mm
e	the second ideal straight plane lateral displacement from the assembled ideal straight plane or band free play (Fig. 5.8)	mm
e_p, e_s	the respective primary and secondary pulleys band free play (Fig. 5.14)	mm
exp_s	secondary pulley groove expansion	mm
f	segment central column width	mm
h	tolerance between the segment hole and its pin	mm
i	speed ratio (secondary pulley radius/primary pulley radius)	-
k	constant	N/m
l	symbol for length	mm
l_w, l_b	distances as defined in Fig. 5.15	mm
l_{bP}	Involute profile distance as defined in Fig. 5.12	mm
l_Δ	segment sliding distance along the compression straight due to belt misalignment	mm

$l_{\Delta_{\max}}$	maximum sliding distance of the segment allowed by the proposed model due to belt misalignment	mm
l_P, l_S	distances as defined in Fig. 7.2	mm
m_b	mass of bands per unit length (the two sets of bands combined together)	kg/m
m_s	mass of segments per unit length	kg/m
n	number of segments clamped along the compression straight	-
ω_P	primary pulley speed	rad/s
ω_S	secondary pulley speed	rad/s
p_m	mean pressure of lubricant between segment and pulley	N/m ²
p_{mean}	average pressure	N/m ²
press_P	primary pulley pressure	N/m ²
press_S	secondary pulley pressure	N/m ²
q	distance of shoulder above rocking edge of segment (Fig. 5.8)	mm
t	lubricant thickness between 2 separated surfaces; band thickness (10 bands altogether)	mm; mm
t_B	band thickness (10 bands altogether)	mm
tetam_P	angle of optical measuring system	deg
torque_P	primary pulley or input torque	N.m
torque_S	secondary pulley or output torque	N.m
v_{rel}	relative velocity between 2 surfaces separated by lubricant	m/s
v_{seg_P}	number of segments passing along the compression straight per second	seg/s
v_{sl_P}	sliding bands speed during exit from the primary pulley	m/s
ω_P, ω_S	primary and secondary pulleys rotational speed respectively	rpm

$x_{aP}, x_{aS},$ x_{bP}, x_{bS}	points or distances as defined in Fig. 5.2 and 5.7 where subscripts P and S refer to the primary and secondary pulley respectively	mm
$\bar{y}$	centre of area distance from the rocking edge of the segment contact area as defined in Fig. 5.10	mm
y_{aP}, y_{aS}	points or distances as defined in Fig. 5.7	mm
A	viscous shear contact area; segment contact area; a constant	$m^2; m^2;$ N
A_{seg}	effective cross-sectional area of the segment	mm^2
C	segments compression force	N
$C_1, C_2,$ C_3, C_4	segments compression forces along the straights, where the subscripts referred to the respected exit and entry points of the primary and secondary pulleys	N
D_1, D_2	pressure forces in the segments on the tight and slack side respectively	N
E_{seg}	segment Young's Modulus of elasticity	N/mm^2
F	total frictional force at the contact; total frictional viscous shear force; axial compression force; force	N
F_B	band force towards centre of shaft (the two sets of bands combined together)	N
$FB_{B12},$ FB_{B34}	sliding frictional forces between the band and the primary pulley cone surfaces during exit and entry into the primary pulley respectively	N
F_P, F_S	primary and secondary pulleys axial compression forces respectively	N
F_{segP}	axial compression force per segment at the primary pulley	N/seg
F'_{segP}	primary pulley axial compression force per segment at exit and entry from tangency point to take of point	N/seg
F_x, F_y	component of forces in the x and y-axes respectively	N

H_P, H_S	input and output power of the primary and secondary pulleys respectively	W
H'_S	actual output power at the secondary pulley (after considering belt power losses)	W
J_P, J_S	the respective primary and secondary pulley pressure	N/m^2
K_f	gradient for the pulley cone radii of curvatures equation	-
K_{f1}	constant for the pulley cone radii of curvatures equation	mm
L	belt length	mm
ML_S	output torque loss	$N.m$
M_P, M_S	the respective input and output torque of the primary and secondary pulleys	$N.m$
M'_S	actual output torque at the secondary pulley (after considering belt torque losses)	$N.m$
N	total normal reaction force at contact	N
N_{BP}	frictional force between the bands and the segments at the primary pulley	N
N_P	reaction force of the segment on the primary pulley cone surfaces	N
N'_P	normal component force on the primary pulley cone surfaces	N
P	force acting on the segment shoulders	N
PL_{bsl}	total power loss due to band sliding against pulley cone surfaces that resulted from excessive belt misalignment	W
$Pl_{ex1}, Pl_{en2}, Pl_{ex3}, Pl_{en4}$	the respective power losses at entry and exit points of the primary and secondary pulleys, due to band sliding against the pulley cone surfaces that resulted from excessive belt misalignment	W
Pl_{exc}	total power loss due to excessive belt misalignment	W

Pl_{sl}	total power loss due to segment sliding during exit and entry	W
$(Pl_{sl})_{Exp},$ $(Pl_{sl})_{EnP}$	total power losses due to segments sliding at the primary pulley during exit and entry respectively	W
Pl_{slP}, Pl_{slS}	power losses due to segments sliding at the primary and secondary pulleys respectively	W
Pl_{sml}	power loss due to small belt misalignment along the compression straight (due to segments sliding against each other on their back and front faces)	W
P_m, P_{m1}, Pl_{m2}	the respective intersection points between the belt mid-points locus during ratio change and the initially assembled ideal straight plane of the primary and secondary pulleys (as defined in Fig. 3.2)	
PP	the locus line that represents the motion of the belt mid-points at the primary pulley	-
Q_{DN}	driven pulley thrust	N
R	belt radius	mm
R_a, R_b	distances as defined in Fig. 5.14	mm
RA_P, RA_S	belt radii to the arbitrary centre of contact area between the segments along the straights for the primary and secondary pulleys respectively	mm
RB_{med}	belt radius at speed ratio $i = 1.0$	mm
RB_{mP}, RB_{mS}	the respective primary and secondary pulley radii at the initially assembled zero belt misalignment or ideal straight plane	mm
$RB_{P1}, RB_{P2},$ RB_{S1}, RB_{S2}	the respective pulley radii as defined in Fig. 3.1 and 3.2	mm
$RC_P, RD_P,$ $RR_P, RBM_P,$ $RBF_P, RCF_P,$ $RDF_P, RBMF_P,$ RRF_P	primary pulley belt radii as defined in Fig. 5.8	mm

$RC_S, RD_S,$ $RR_S, RBM_S,$ $RBF_S, RCF_S,$ $RDF_S, RBMF_S,$ RRF_S	secondary pulley belt radii as defined in Fig. 5.8	<i>mm</i>
R_{DR}	driven pulley radius	<i>mm</i>
S	tensile force in the bands	<i>N</i>
SS	the locus line that represents the motion of the belt mid-points at the secondary pulley	-
S_m, S_{m1}, S_{m2}	the respective intersection points between the belt mid-points locus during ratio change and the initially assembled ideal straight plane of the primary and secondary pulleys (as defined in Fig. 3.2)	-
T	band tension (the two sets of bands combined together)	<i>N</i>
T_1, T_2	tension in the bands on the tight and slack sides respectively	<i>N</i>
$T_1, T_2,$ T_3, T_4	band tensions (the two sets of band combined together), where the subscripts refer to the respective exit and entry points of the primary and secondary pulleys	<i>N</i>
T_{in}^*	maximum transmitted torque	<i>N.m</i>
TPL	total power loss of the belt	<i>W</i>
U_{IP}	work done per segment as the segment travels from the tangency point to the point where it recovers its full free length	<i>N.m/seg</i>
U_{BP}, U_{segP}	band and segment centrifugal forces respectively	<i>N</i>
U_{seg}	work done per segment along the compression straight	<i>N.m</i>
W_P	frictional force between primary pulley cones and segments sides surfaces	<i>N</i>

X	centre distance between the primary and secondary pulley shafts	<i>mm</i>
X'	clamping distance of the segments along the compression straight	<i>mm</i>

ABBREVIATIONS

App.	appendix
calc.	calculated
CVT	continuously variable transmission
IVT	infinitely variable transmission
ehd	elastohydrodynamic
eqn.	equation
Expt.	experiment
Fig.	figure
meas.	measured
SF	safety factor
o/d	overdrive
u/d	underdrive

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CHAPTER 1

INTRODUCTION

1.1 Introduction

A vehicle that is powered by an internal combustion engine will be quite impossible to be driven smoothly and comfortably without the aid of a transmission system. This, by itself explained the purpose and importance of a transmission unit in a vehicle. That is, they are the components that made it possible to transfer the power or torque from the engine to the driving wheels. Giles J. G. 1969, described the vehicle transmission as the unit that is required to transfer the power developed by an engine to the driving wheels so as to propel the vehicle. Though in some literature, the transmission was called by a different name, for examples, the Automotive Handbook by Bosch, Germany, 1993, called it drivetrain and Bader C. 1990 and Deacon M. et al 1994, called it powertrain (however, these terms usually include the engine as well), it all means the same thing and carries out the same function. In brief the author would say that, the primary function of a vehicle transmission is to perform the above mentioned task and its secondary functions are to provide the required vehicle acceleration or tractive effort at all vehicle speeds and at the same time if possible, to improve fuel (energy) consumption as well.

1.2 Continuously Variable Transmission (CVT)

1.2.1 Introduction

At the present time, with the stricter rules and regulations on environmental pollution, the demand for 'greener' and more environmental friendly vehicles and competition among vehicle manufacturers themselves, automotive engineers are faced with a very challenging future. With the technology having nearly reached saturation level, they still have to come out with a better performance, low fuel consumption, cheaper and more efficient vehicles. Hence, in facing these endless competitive battles for a better and safer future, automotive transmission engineers have to make ingenious designs and developments to improve transmission performance.

One of these designs is the Continuously Variable Transmission (CVT) or Infinitely Variable Transmission (IVT). Beachley N. H. and Frank A. A. 1980, defined CVT as a transmission that has speed ratio that can be varied continuously over its allowable range while IVT (basically the same as CVT) with the added restriction that a speed ratio of zero must be available. To achieve both low fuel consumption and good vehicle performance, usually one has to sacrifice one or the other. That is, for low fuel consumption, one has to operate the transmission along the ideal operating line of the engine fuel map while to attain high performance, one needs a high engine torque and low gear ratio, if possible still operating the engine along the ideal operating line. These are possible with a CVT or IVT.

Thus, CVT or IVT has always been the dream for all transmission designers and engineers. This is because, CVT or IVT has the capability and ability to fulfil all the above mentioned functions that are required by a vehicle transmission.

1.2.2 Brief History

It all started in the early days of the motor industry, way back in 1886 where Benz and Daimler equipped their first mass produced gasoline-engine-powered vehicle with a rubber V-belt CVT [Lowenthal S. H. et al 1983]. However, Daimler Benz abandoned their belt drive in 1903 leaving the French manufacturer Fouillaron which used V-belt variable cone pulley CVT. Fig. 1.1 shows one of the early designs of variable cone pulley CVT. By 1911, belt-drives were 'practically' obsolete for automotive use. George S. Strong 1903, introduced variable stroke transmission CVTs which used hydraulic and reversible roller ratchet clutches operated intermittently and was incorporated into Union Trucks in Philadelphia (Fig. 1.2 and 1.3), while in 1905, the Newman's manually shifted variable stroke CVT which changed the drive stroke ratio, was demonstrated in a Daimler Car at the British Olympia [Gott P. G. 1991].

In 1909, Cartercar, Metz, Sears, Camber, Petrel and Worth were among the early manufacturers using friction drives CVTs that were composed of single or double-discs. Fig. 1.4 and 1.5 show single and double disc design principles adopted by these manufacturers. In Great Britain around the same time, T. W. Barber introduced his

hydraulic control variable stroke transmission and was applied to Hutton automobile. In 1924, hydrostatic drives CVTs were introduced by Automatic Clutch and it was installed in a Buick. In 1925, Mssr. Sensaud de Lavaud introduced a variable throw CVT using a wobble plate in conjunction with roller ratchet drives on the rear axle which Buciali used in their Tan six- and eight-cylinder front wheel models (Fig. 1.6). Another variable stroke that was developed and used was the R.v.R. variable stroke torque converter (Fig. 1.7), it was fitted into the crankcase of a Minerva by the Belgian firm of Minerva where, each of the engine crank pins was connected via a triangular rocking member to its own roller ratchet. The variable throw and hence its variable speed ratio was achieved by moving the triangular pivot point. [Gott P. G. 1991]. In 1935, the British Austin Sixteen were equipped with a double toroidal traction drive developed by Buick Division of General Motors. The design was similar to that of the Hayes Self-Selector transmission patented in 1929 [Lowenthal S. H. et al 1983].

In 1958, Dr. Hub van Doorne, founder of the Van Doorne Transmissie introduced the 'Variomatic' rubber V-belt CVT. It was initially applied to low-powered cars only, but as a result of new development on the rubber V-belt, it was then used in cars up to 1.4 litre with 51 kW of power. Since then, under the name of DAF and from 1975 under the name of Volvo (Volvo 340 series) it is claimed that, 1.2 million cars have been equipped with the 'Variomatic' [Hendriks E. et al 1988]. In 1985, Fuji Heavy Industries mass produced the metal pushing V-belt CVT, the 'Transmatic' that was introduced by the same person, Dr. Hub van Doorne, which Subaru started using in its Justy Model. A 30% improvement in fuel economy is claimed over their previous automatic transmission [Volz T. A. 1989].

Although there were and are other outstanding CVT designs, that have been tried and applied for automotive applications, the author did not include these in this work, since this is not a History report. However, the reader is referred to Gott P.G. 1991, for more CVTs information.

1.3 Metal Pushing V-belt CVT

1.3.1 Introduction

The Van Doorne metal pushing V-belt CVT (Fig. 1.8) comprises basically a pair of variable vee-shaped pulleys (secondary and primary) and an endless composite metal belt. The endless metal pushing V-belt (Fig. 1.9) is wound around both the pulleys (Fig. 1.10). The input pulley which is usually called the primary pulley is driving the transmission (hence sometimes called the driving pulley) while the output or secondary pulley is being driven (hence sometimes called the driven pulley), which in turn is connected to the vehicle wheels through the final drivetrain. The pulleys are constructed in such a way that, one-half of each pulley is fixed to the shaft while the other halves are adjustable through sliding along their respective shafts. The axially adjustable vee-shaped pulleys enable the metal pushing V-belt to slide radially outwards and inwards and thus the effective belt radius can be steplessly adjusted (Fig. 1.10).

Generally, the control is designed in such a way that the primary pulley determines the speed ratio while the secondary pulley, ensures that the proper belt tension is always maintained to prevent slipping [Mott P. J. 1993 and Morimoto Y. 1993].

1.3.2 Continuously Variable Speed Ratio

Figure 1.10 depicts clearly how the variable speed ratios of the composite metal pushing V-belt CVT are achieved. Simply, when the endless and inextensible composite metal pushing V-belt is being squeezed by the input primary pulley, it will then be forced to slide radially outwards on the vee-shaped pulley. Thus, increasing the effective primary pulley belt radius. Since the belt tension has to be maintained taut at all times by the secondary pulley, the converse happens at the other end of the belt. The adjustable half of the secondary pulley will be made to slide axially away from the fixed half as the composite metal pushing V-belt slides radially inwards on this vee-shaped pulley, thus decreasing the effective secondary pulley belt radius.

Alternatively, when the primary pulley pressure is reduced, the composite metal pushing V-belt CVT will slide radially inwards reducing the primary pulley effective

radius. Conversely, due to the requirement of maintaining the appropriate belt tension as explained earlier at the secondary pulley, the secondary pressure with the aid of its internal spring will squeeze accordingly the composite metal pushing V-belt radially outwards, thus increasing the secondary pulley effective radius.

When the effective radius of both the pulleys are equal, a one to one ratio will be achieved. An underdrive (speed ratio reduction) or low gear is achieved when the effective primary pulley radius is less than the effective secondary pulley radius. Conversely, overdrive or high gear is achieved when the effective primary pulley radius is greater than the effective secondary pulley radius. For the Van Doorne metal pushing V-belt CVT used in this work which was taken from a Nissan Micra 1.3 litre engine car (engine type: CG13DE), its speed ratio range is approximately 2.5037:1 low gear to an overdrive of 0.4977:1 (Fig. 1.10).

1.3.3 Design

Figure 1.9 shows the general design and Fig. 1.11 shows the design of the standard type and new generation type segments of the Van Doorne metal pushing V-belt. It comprises basically of 2 sets of thin metallic band strips and a number of thin metallic plates trapezoidal in shape usually called segments or blocks. The segments are sprung together by the 2 sets of bands through the segments location slots. These slots allow the segments to slide freely along the sets of bands. The bands are made from maraging steel with high fatigue strength and the segments are made from roller bearing steel. Currently, the number and size of segments and the number of band strips determine the metal pushing V-belt power capacity class. There are now a range of belt configurations, available from the Van Doorne Transmissie belt manufacturer [Hendriks E. 1993].

Fig. 1.12 shows the general spec of the Van Doorne metal pushing V-belt CVT used in this work. The belt has about 280 segments, each segment being approximately 2 mm thick, 24 mm wide and 14.5 mm deep, and the total endless belt length was measured to be approximately 604 mm, and 2 sets of 10 band strips, the band thickness averaging 0.2 mm thick and 8.65 mm wide.

1.3.4 Power Transfer Design Concept

It is here that researchers, differ in their opinion on how actually the metal pushing V-belt CVT transmits power from the primary to the secondary pulley. To the best of the author's knowledge, there are only two school of thoughts on this matter. The first school of thought, that is researchers like Cuypers M. H. 1980, Gerbert G. 1984, Becker H. J. 1987, Hendriks E. et al 1988, Sun D. C. and Watson T. J. 1988, Fuji T. et al 1993a and 1993b, predicted that power is transferred through Coulomb friction while, the second school of thought of researchers like Micklem J. D. et al 1994b, based theirs on viscous shear and elastohydrodynamic (ehd) theory.

Basically, the composite metal pushing V-belt transfers its power from the primary to the secondary pulley through the metal belt. When the segments are squeezed together by their respective pulley, contacts occur through the wedging action between the segment sides and the pulley cone surfaces. Thus, in drive conditions, the primary pulley will force the segments through these contacts in the direction of rotation of the primary pulley and at the same time forcing them out radially. Since the segments are held and guided together by the two inextensible and flexible sets of bands, torque is transferred through the contacts from the primary pulley cone surfaces to the segment sides. The segments are then forced and pushed against each other (front to back) making contact with each other and slide along the two sets of bands. The locating pin and slot of the segment help to ensure that the segments are always in transverse position to the set of bands.

At the secondary pulley, the converse takes place, torque will be transferred from the segments to the pulley through the contacts between the segment sides and the pulley cone surfaces. Between these two contacts on the primary and secondary pulleys, the composite metal pushing V-belt resembles a continuous set of struts that are being compressed together at both ends. Along the return side of the metal pushing V-belt CVT, the segments will be transported back by the two sets of bands to the primary pulley and the process is then repeated. Thus, power is being transmitted by compression instead of tension as in the conventional belt.

1.3.5 Coulomb Friction Theory

As has been explained in the previous section, the composite metal pushing V-belt CVT, transfers its power from the primary pulley to the belt and from the belt to the secondary pulley through the interfacial contacts between the segment sides and the pulley cone surfaces. That is, the frictional forces are being transmitted through these contacts. The author would summarised the main force flow of the metal pushing V-belt CVT as follows:

- i. the torque from the transmission input shaft, flows through to the primary pulley via the shaft's splines ,
- ii. the torque then flows to the segments through the interfacial contacts between the primary pulley cone surfaces and the segment sides,
- iii. as a force now, it then flows through each segment by pushing against each other forward, via their interfacial line contacts or rocking edges (while on the active section of the primary pulley) and surface contacts (while on the straight section between the pulleys along the two sets of bands),
- iv. the force then flows to the secondary pulley through the interfacial contacts between the segment sides and the secondary pulley cone surfaces converting itself back as a torque and finally,
- v. the torque then flows to the output transmission shaft via a similar way as that of the primary pulley.

The main difference between the two schools of thoughts, regarding how to actually mathematically model the force flow through the metal pushing V-belt CVT, lies at the above mentioned interfacial contacts of the transmission. The first school of thought of researchers as those mentioned earlier (section 1.3.4) assumed that, at these interfacial contacts, Coulomb friction exists. That is,

$$F = \mu \cdot N$$

By assuming Coulomb friction, Gerbert G. 1984, Becker H. J. 1987, Sun D. C. and Watson T. J. 1988, extend the classical V-belt theory to include the functions of both the tension and compression of the belt elements.

1.3.6 Viscous Shear And Elastohydrodynamic Theory

The second school of thought of researchers [Micklem J. D. et al 1994b] assumed that, at the lubricated interfacial contacts, viscous shear exist. This leads them to model mathematically the force flow through the metal pushing V-belt CVT using viscous shear and elastohydrodynamic lubrication theory. The viscous shear equation is represented by

$$\tau = \eta \cdot \frac{V_{rel}}{t}$$

With this assumption, frictional shear force transferred through the interfacial contact is obtained by multiplying the viscous shear stress with the viscous shear contact area. That is,

$$F = \tau \cdot A = \eta \cdot \frac{V_{rel}}{t} \cdot A$$

Thus with this assumption, the frictional force is now a function of the lubricant viscosity and thickness, relative velocity of the surfaces and the actual area of contact. Where as, in the Coulomb friction model, the frictional force is a function of the normal reaction force and the coefficient of friction between the contact.

However, with viscous shear and elastohydrodynamic lubrication, all the variable quantities mentioned above, are quite difficult to measure and obtain to a reasonable accuracy. The lubricant viscosity, is itself a function of the lubricant pressure, temperature and shear rate (except for Newtonian fluid). The lubricant thickness on the other hand, requires advance technology to be measured to a reasonable accuracy. Again, the relative velocities between the pulleys and the metal pushing V-belt CVT, are also not easy to measure. These, probably lead Micklem J. D. et al 1994b to assume according to elastohydrodynamic lubrication theory the transmission lubricant viscosity and thickness. The relative velocities of all the moving parts of the belt, were calculated from their own mathematical expressions.

All these, the author would say, has led Micklem J. D. et al 1994b to make further assumptions to justify their model. In their paper on “Modelling of the Steel Pushing V-belt Continuously Variable Transmission” itself, contains not less than 10 further assumptions. One of these, is to assume that, the viscosity of the lubricant between the

segment sides and the pulley cone surfaces obeys the following simplified equation, generally called the Barus Law [Gohar R. 1988]. That is,

$$\eta_s = \eta_{os} \cdot e^{\alpha_s} \cdot p_m$$

From the author's knowledge of elastohydrodynamic lubrication theory, this equation itself was originally developed based on a few other assumptions and has some limitations; for example, it is applicable to Newtonian fluid only [Gohar R. 1988]. However, these are part and parcel of the life as an engineer. As has been mentioned by Professors of Mechanical Engineering, Juvinall R. C. and Marshek K. M. 1983 in chapter one of their *Fundamentals of Machine Component Design* book (second edition), "Scientists do not completely understand electricity, but this does not prevent electrical engineers from developing highly useful electrical devices; similarly, scientists do not completely understand combustion or metal fatigue, but mechanical engineers use the understanding available to develop highly useful combustion engines". Thus, this is where engineers use their engineering judgement to obtain solutions to practical problems.

1.4 Discussion

The brief explanation in the previous sections, leads to quite a number of questions to be answered. These, require research and development work to be carried out on the Van Doorne metal pushing V-belt CVT to obtain answers for the queries raised and at the same time improving its performance. The author's main questions are, "How does the metal pushing V-belt CVT actually transmit power?" and "Which theory or model should be used to really represent and simulate the actual behaviour of the transmission?". It does not seem that the metal pushing V-belt CVT could be represented by a straight forward mathematical formula as in a conventional belt transmission. Why the author says this is because of the nature of the Van Doorne metal pushing V-belt CVT itself. It is made up of discrete components which are not tied to one another. That is, they can move relative to each other and thus it will be very difficult to mathematically model their performance to a reasonable accuracy. These relative movements between the metal pushing V-belt components will probably lead to slipping and hence will be accompanied by power loss and this in

turn will affect the transmission efficiency and performance. Some of them are as follows:

- i. The relative movement of both the pulleys sliding halves with respect to their fixed halves during speed ratio change which give rise to belt misalignment.
- ii. The relative radial movement of the belt with respect to the pulleys during speed ratio change.
- iii. The relative transverse movement of the segments themselves with respect to each other along the straights due to belt misalignment.
- iv. The relative tangential sliding movement of the segments in driving conditions with respect to the pulley circumferential speed at both pulleys.
- v. The relative lateral and transverse movement of the segments in the driving condition with respect to the first or inner bands (remember there are 2 sets of bands).
- vi. The relative movement of the segments on the compression and slack side in driving conditions with respect to each other. (Note: Both these relative movements can result in noise and vibration).
- vii. The relative lateral and transverse movements of the bands with respect to each other.
- viii. The relative rotational movement of the segments about the locating pins with respect to each other during operation.
- ix. The relative rocking movement of the segments about the rocking edges with respect to each other during operation.

These are just a few examples of relative motions that exist in the transmission, there are probably many more. Of course, some of the above relative motions have negligible or probably no effect at all to the transmission efficiency and performance. For example, the one due to belt misalignment effect is assumed by Sun D. C. and Watson T. J. 1988, to be small for the convenience of his analysis, which the author intends to investigate in this work. Nevertheless, from the engineering perspective, whatever is the case, all these relative motions contribute, if not much, at least to some degree of power loss. For example, if each one of the above relative motions,

contributes only 0.5 % power loss, it means that, the total power loss can amount to about 5 % of the total power.

As has been mentioned earlier, there are basically two schools of thought in modelling and predicting the performance of the metal pushing V-belt CVT,

- i. Coulomb friction or
- ii. viscous shear and elastohydrodynamic lubrication theories.

In the Coulomb friction model, it was assumed that most of the power is being transmitted due to friction force between the contacts of the segment sides and the pulley cone surfaces. Beachley N. H. and Frank A. A. 1980, further assumed that the bands also contribute a small amount of power as well. They based this prediction on the assumption that, the compression forces between the segments should be less than the bands tension to prevent kinks; hence, some power will be transmitted due to this net force between the bands tension and segments compression.

In the viscous shear and elastohydrodynamic lubrication model, it was assumed that most of the power is being transmitted via viscous shear forces between the contacts of the segment sides and the pulley cone surfaces. Viscous shear was also predicted at the contacts between the first or inner bands and the segments shoulders and between the adjacent bands [Micklem J. D. et al. 1994b].

Based on the literature study that has been carried out, majority of the work done on the metal pushing V-belt CVT were based on Coulomb friction. Experimental investigations by various researchers have correlated well with Coulomb friction theory (Chapter 2). Furthermore, it is not the intention of the author to dwell into the concept of viscous shear and elastohydrodynamic lubrication or Coulomb friction theories, but to investigate the design and performance of the Van Doorne metal pushing V-belt CVT with regards to its belt misalignment that will probably give rise to friction losses. Thus, an acceptable engineering tool that can be validated experimentally will be required to model the behaviour of the Van Doorne metal pushing V-belt CVT in order to carry out the intended work. Since, Coulomb friction

theory is an acceptable engineering concept for predicting and investigating work involving friction, the author feels that it will be wise to choose Coulomb friction theory for this work, and probably use viscous shear and elastohydrodynamic lubrication theories as future work.

Viscous shear and elastohydrodynamic lubrication theories on the other hand, has only been applied recently and it is being used by a group of researchers only, that is Micklem J. D. et al 1994b. Thus, this made the author feels that it probably requires further work and justification which the author has no intention to dwell into. Furthermore, the experimental equipment that is required to justify and validate its theoretical results are of the high end products which are very expensive and beyond the budget of this project. Hence, this should probably justify the author for not choosing viscous shear and elastohydrodynamic lubrication theories for this work.

1.5 Objectives And Scope

1.5.1 Objectives

The main objective of this research is to carry out a research study on the effect of belt misalignment of the Van Doorne metal pushing V-belt CVT for automotive applications with regards to its efficiency. It was planned for this research study to produce new information in predicting the motion of the belt elements.

1.5.2 Scope

The research contains both theoretical and experimental investigation. Measurements were carried out and theoretical prediction will be made to aid in investigating the misalignment and movement of the belt elements. A test rig constructed to use with the Van Doorne metal pushing V-belt CVT and an a.c. electrical motor capable of transmitting up to 9 horse power (approximately 6,700 watts). Then, an experimental test was devised to measure the movement of the belt elements and correlation studies between the theoretical and experimental values carried out.

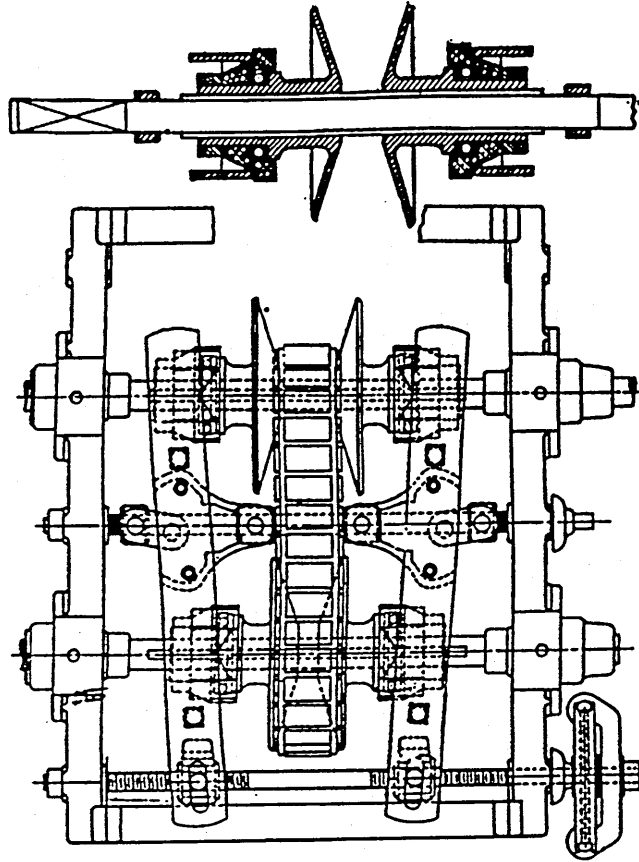


Figure 1.1 Variable cone pulley CVT [Gott P. G. 1991]

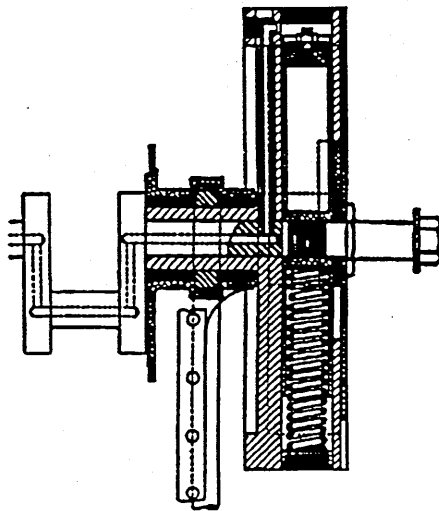


Figure 1.2 Cross-sectional view of the Strong variable throw mechanism in an engine flywheel. Output arm is in neutral position at the centre of the flywheel [Gott P. G. 1991].

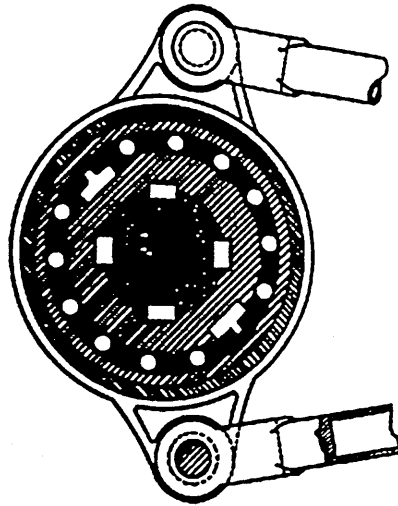


Figure 1.3 Roller ratchet used on the rear axle with the Strong transmission [Gott P. G. 1991].

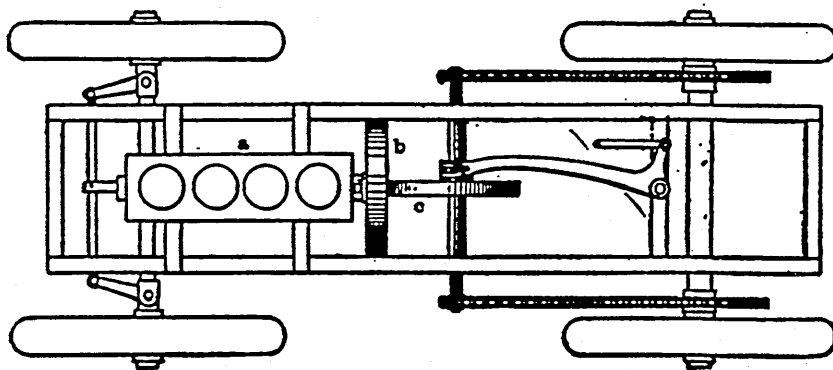


Figure 1.4 Single-disc change gears [Gott P. G. 1991].

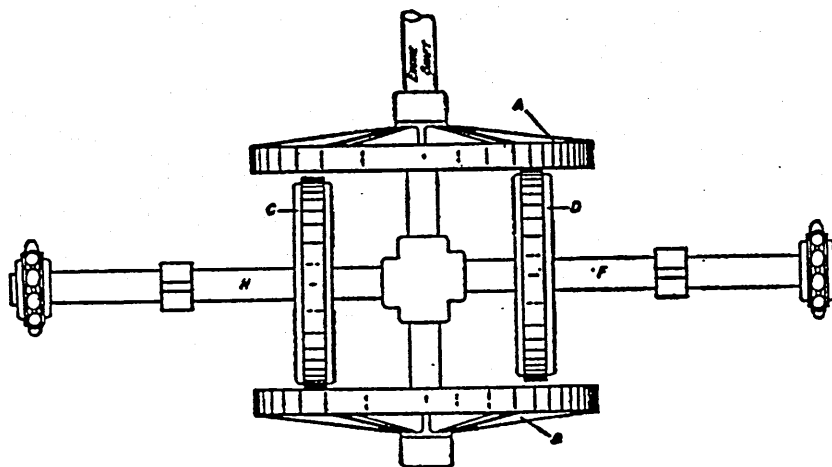


Figure 1.5 Double-disc, double wheel change gears [Gott P. G. 1991].

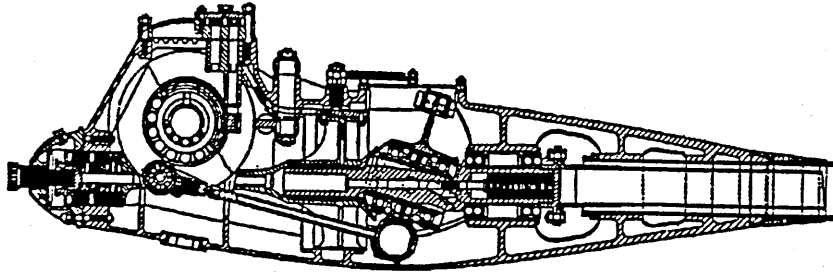


Figure 1.6 De Lavaud variable stroke CVT [Gott P. G. 1991].

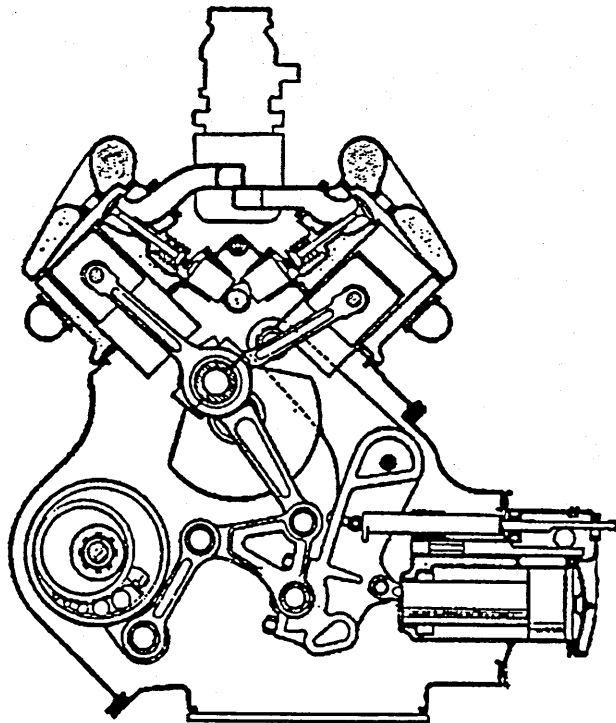


Figure 1.7 R.v.R. variable stroke torque converter fitted into the crankcase of a Minerva V-8 [Gott P. G. 1991].

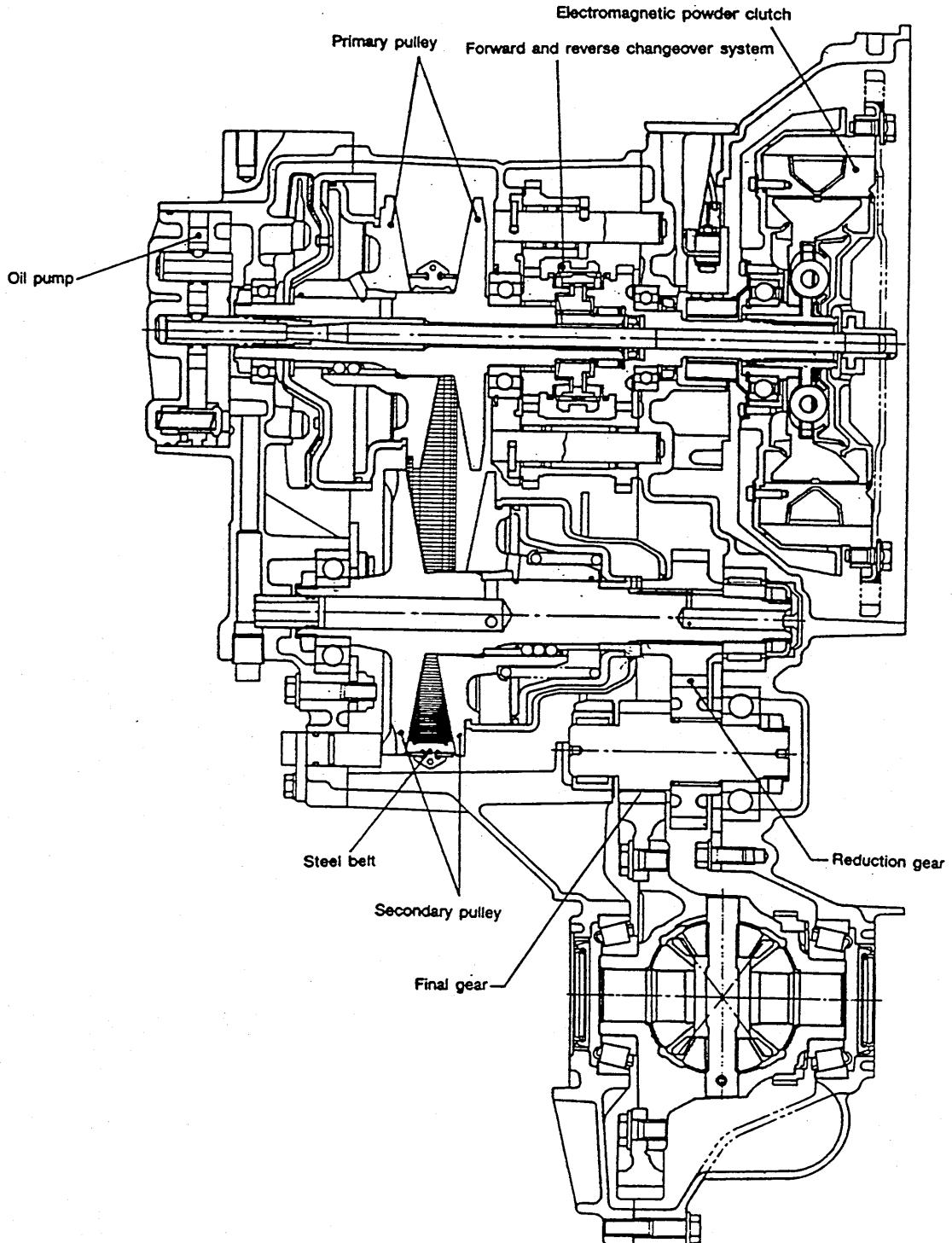


Figure 1.8 Section view of a transverse Van Doorne metal pushing V-belt CVT for the NISSAN MICRA 1.3L CVT [NISSAN MICRA CVT workshop manual].

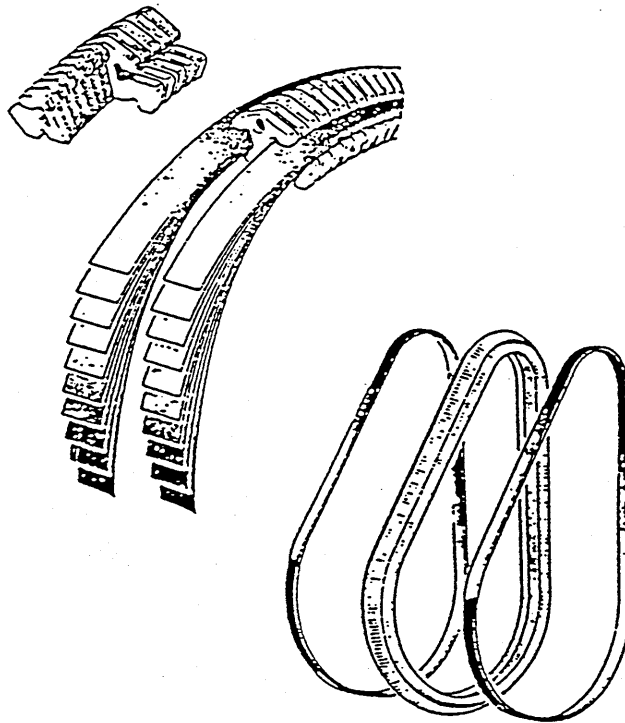


Figure 1.9 Van Doorne metal pushing V-belt, consisting of approximately 300 segments and 2 sets of 10 bands [Hendriks E. 1993].

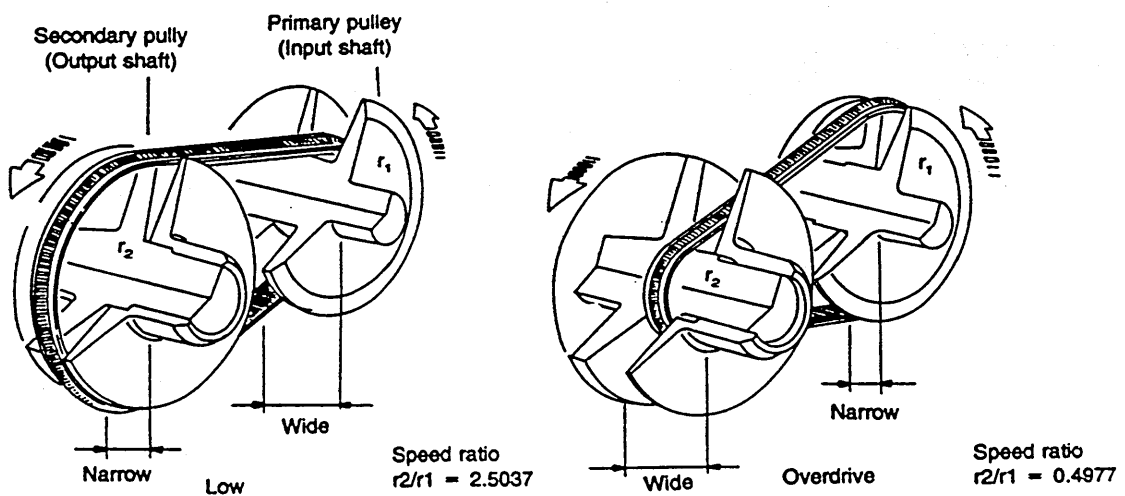


Figure 1.10 Illustration of pulleys and belt under and overdrive speed ratios [NISSAN MICRA CVT workshop manual].

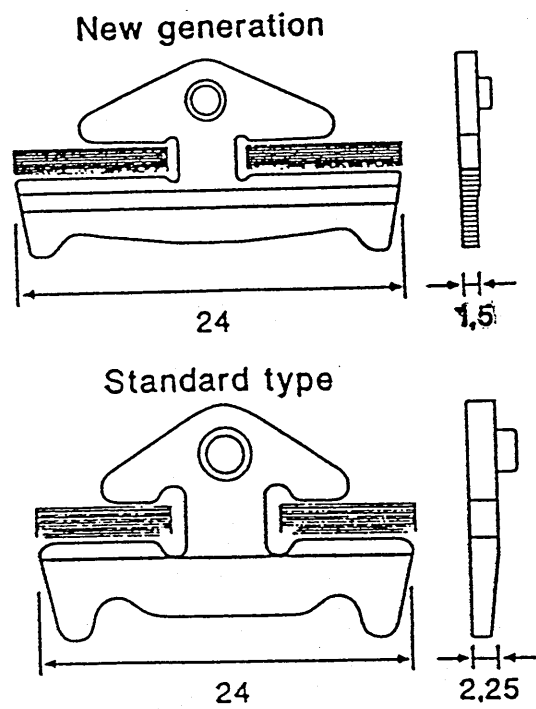


Figure 1.11 Standard type and new generation belt [Hendriks E. 1993].

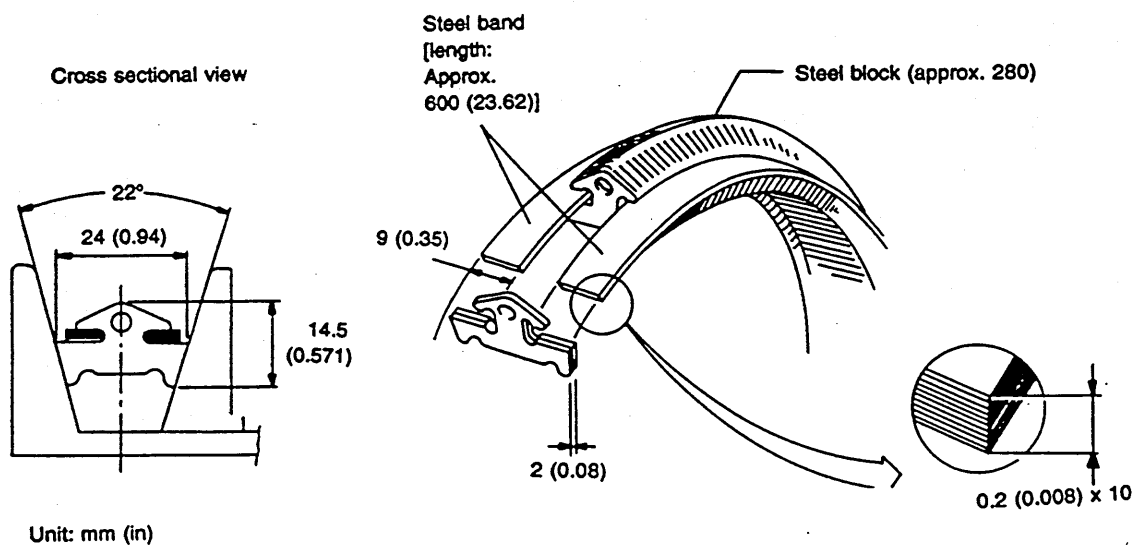


Figure 1.12 Van Doorne metal pushing V-belt, consisting of approximately 280 segments and 2 sets of 10 bands [NISSAN MICRA CVT workshop manual].

CHAPTER 2

LITERATURE REVIEW

2.1 Introduction

Most literature published on metal pushing V-belt CVTs for automotive applications usually discussed the belt only in general terms. It's history, advantages and disadvantages, performance and limitations, its future use and potentials, but very few really went into the engineering aspects such as, the mechanical and design aspects of the metal pushing V-belt itself. However, the author will review the latter type of publication only. Also, this literature review will focus on two major issues that interest the author on the metal pushing V-belt CVT. That is, how researchers analysed power transfer and how they represent frictional forces within the metal pushing V-belt CVT components. In general, most researchers extend the theory of classical V-belt in predicting the metal pushing V-belt performance and used Coulomb friction theory to model the frictional forces that are transferred within the metal pushing V-belt CVT. However, there is one group of researchers that use viscous shear and elastohydrodynamic lubrication theory to model the frictional forces.

2.2 Literature Review

Among the pioneer researchers involving themselves with the engineering aspects of metal pushing V-belt CVTs were Swain J. C. et al 1980, who conducted a design study and performance analysis on a steel V-belt CVT for electric vehicles. They tried to develop the steel V-belt concept into a preliminary design of a functional CVT model and coupled it to a high speed output shaft of an energy storage flywheel of an electric vehicle. Their maximum theoretical tension ratio was approximately based on equation,

$$\frac{T_1}{T_2} = e^{\frac{\mu_\beta}{\sin \theta}} \quad (2.2.1)$$

where, T_1 is the band tension on the tight side while T_2 is on the slack side. The author assumes this contains a typing error of μ_β instead of $\mu\beta$. This indicates that Swain J. C. et al assumed that power is being transferred through Coulomb friction

between the belt and pulleys. From their study, they claimed that the results were encouraging with regards to weight, size and efficiency. They found that the efficiency of the transmission was a function of the speed ratio and the amount of power being transferred, and achieved an average experimental efficiency value of about 94.5 %.

Another pioneer researcher involving himself with metal pushing V-belt CVTs is Cuypers M. H. 1980. He was regarded by Horowitz A. 1980 as the one who started the theoretical and experimental work on CVTs; his initial work and initiative at the Department of Mechanical Engineering of the Eindhoven University, Holland was soon indicated as by far the most promising starting point for any high grade CVTs; and consequently, all effort were concentrated on the development of high grade V-bands (belt or chains) suitable for CVTs. Cuypers M. H. predicts that power transmitting of CVTs is fundamentally based on friction and he divided them into 2 main groups:

- i. V-belt variable speed drives which consist of at least one flexible friction body such as belts and chains.
- ii. Traction drives which consist of rigid friction bodies such as balls, cones, rings, toroids and discs.

Cuypers M. H. further subdivided both groups into oil lubricated and oilless transmission and predicts that the transmission of power for the metal pushing V-belt CVT could be based on the following expression,

$$\frac{S - D_1}{S - D_2} \leq \mu = e^{\frac{\mu \beta}{\sin \theta}} \quad (2.2.2)$$

Centrifugal forces has been neglected and the tensile force S in the bands was assumed to have a constant value and it is always greater than the compression forces D_1 and D_2 . Gerbert B. G. 1984, also shared a similar assumption in considering the mechanics of the metal pushing V-belt CVT.

Horowitz A. 1980, as has been mentioned earlier is also among the pioneer researchers involving himself with metal pushing V-belt CVTs. However, most of his data originated from Cuypers M. H.'s work. He evaluated and compared some novel CVTs (including metal pushing V-belt CVT) with current variable and fixed drive

transmissions. From his findings, he found that metal pushing V-belt CVT is the most potential and superior CVT with an approximate overall efficiency at half load of 96%, as compared to Super V-chain and Combi V-chain CVTs with only 92% and 94%, respectively. He also forecasted that, the metal pushing V-belt CVT concept will finally lead to CVTs with high efficiency applicable for any power rating.

In 1984, Gerbert B. G. developed the metal pushing V-belt mechanics based on Coulomb friction from his previous work (Gerbert B. G. 1972 and 1981) on rubber V-belt. To the author, it has been one of the most extensive works on the metal pushing V-belt mechanics, where mounting and initial tension, transmission torque, play in the belt, transition zone and a few others have been discussed in quite detail. However, his assumption of the metal pushing V-belt consists of wedge-shaped plates (segments) which are “frictionlessly hinged” on flexible endless bands is probably too optimistic. By regarding the belt segments as “frictionlessly hinged” on the endless bands means that the coefficient of friction between the segments and the bands is zero. This is of course the most ideal case and could never happen in real life. The implication of assuming that the coefficient of friction between the segments and the bands equals to zero means that, the band tension will have to be constant. That is, the bands will slide freely on the segments shoulders or vice versa with no energy loss and the bands do not transmit any power. Thus, all the power from the primary to the secondary pulley will be transmitted only through the segments pushing each other forward via their interfacial contacts between the segment sides and the pulley cone surfaces. Thus, this makes the metal pushing V-belt CVT analysis very similar to the classical V-belt where, torque is being transferred through segment compression while, in classical V-belt through the belt tension.

The assumption made by Becker H. J. 1987, is quite similar to the one made by Cuypers M. H. 1980. He assumed that the coefficient of friction around the belt circumference is constant and the bands are being regarded as a single belt with a uniform load on the circumference of the bands. However, with regards to this paper, the author would tend to agree with Micklem J. D. 1990 that Becker H. J.’s free body diagrams on the belt mechanics were incomplete. That is, he has no complete diagram

for a segment. Thus, he has no equilibrium about the shaft axis and hence, introduces error in the corresponding equations.

Another group of researchers assuming Coulomb friction theory to analyse the metal pushing V-belt CVT mechanics is Carvajal J. M. et al 1987. Their paper presented an analysis of the belt in discrete form using a computer program. The paper however, to the author is quite difficult to follow because the notations used were not declared clearly and their free body diagrams (reproduced here as Figures 2.1 and 2.2) are confusing. Fig. 2.2a does not seem to be in equilibrium, and in their paper's section 2.1b (on belt equilibrium, reproduced here as Fig. 2.3) the friction loads X_L and Y_L which are supposed to be determined on the contact arcs are missing from the mentioned figure.

Sun D. C. and Watson T. J. 1988 paper on performance analysis of the metal pushing V-belt CVT is quite interesting. This is because, some of the assumptions made were quite trivial. The one that interested the author most is their assumption (8), that there is no misalignment of the two pulleys. This however, is being followed by a statement saying that in practice, misalignment can hardly be avoided during ratio change operation and the presence of misalignment affects the belt efficiency significantly because it introduces additional power loss, arising from the relative sliding between the segments as they travel on the tangent part of the metal belt, but its effect on the quantities calculated in their paper are expected to be small. This, to the author is quite a contradicting statement which was also highlighted by Micklem J. D. 1990. The same goes for their assumption (5), where the belt active arcs β in their analysis were assumed to have only one value which is quite inappropriate. This is because, it is quite obvious that the active arcs will vary as the speed ratio changes. Furthermore, in their abstract they specifically mentioned that the model that they constructed and analysed takes into consideration speed ratio change. Also in their introduction, they commented that Gerbert B. G. 1984's work (which they considered as the most extensive work) did not treat the speed ratio change effect, which is of crucial importance to the control of the metal pushing V-belt CVT operations. They claimed that their paper addressed these features but in fact they did not. However, in

assumption (6) they revealed that lubrication conditions at the three different interfaces were different and they introduced three different coefficients of friction. That is, coefficients of friction at the,

- i. boundary regime between pulleys and segments,
- ii. hydrodynamic regime between bands and
- iii. mixed regime between segments shoulders and adjacent bands.

These, have similarities with the work of Micklem J. D. 1990, Micklem J. D. et al 1991 and Guebelli M. et al 1992, except that they predicted normal viscous shear stress τ , at all the mentioned interfaces as a function of the oil film thickness t , and is given by the following equation,

$$\tau = \eta \cdot \frac{v_{rel}}{t} \quad (2.2.3)$$

and assumed that the oil film thickness t , is inversely dependent on the average pressure p_{mean} , that is,

$$t = \frac{k}{p_{mean}} \quad (2.2.4)$$

The forces involved were then calculated based on the general formula, force equals stress multiply by area.

A paper by Hendriks E. et al 1988 from Van Doorne Transmissie B. V. itself was probably the first paper, reviewing the Van Doorne metal pushing V-belt CVT history and revealing some of its design features. The one that interests the author most from this paper is the section on external belt geometry, regarding the misalignment problem of the belt which is the main subject of this research work. In this paper, Hendriks E. et al introduced an approximate equation to calculate belt misalignment of the Van Doorne metal pushing V-belt CVT. Unfortunately, they did not show how the equation was derived. As a start for this work, the author has succeeded in introducing an ideal equation to calculate the belt misalignment of the Van Doorne metal pushing V-belt CVT, which was derived geometrically and justified by using Unigraphics Solid Modeller. As claimed by Hendriks E. et al, the author found that his approximate equation to calculate the belt misalignment of the Van Doorne metal pushing V-belt CVT is only accurate for a certain speed ratio range only. The reader is referred to chapter 3 for these findings.

Paper by Roovers G. 1991 (also from Van Doorne Transmissie B. V.) only reviewed very briefly the metal pushing V-belt CVT history and belt efficiency with regards to its design parameters such as slip, friction, dynamic and viscous losses. In this paper, he also revealed the new claimed optimised belt design and the range of belt differentiation scheme available from the company. That is, from a minimum applicable torque of $80 \text{ N.m} / 7500 \text{ rpm}$ to $300 \text{ N.m} / 6500 \text{ rpm}$ (reproduced here as Table 2.1).

Kim H. and Lee J. 1991, in their analysis of the belt behaviour and slip characteristics for metal V-belt CVT also used Coulomb friction theory. Their analysis are similar to that of Gerbert B. G. 1984's work. The forces assumed to act on the metal pushing V-belt elements in the active arc are similar and they also assumed that the band tension is constant which as the author has mentioned earlier applies to ideal case or situation only. In this paper, they revealed the presence of the radial belt displacement around the pulleys which they justified with their theoretical calculations. This finding was then used by Micklem J. D. et al 1994a, to mathematically model the belt torque losses. Kim and Lee also revealed that, the gross slip points of the metal pushing V-belt depend on the torque transmission capacity of the driven side, and they found that the practical coefficient of friction values were in the range of 0.08 to 0.13.

In 1991, the author would say Micklem J. D. et al revolutionised the modelling of power transfer concept of the metal pushing V-belt CVT by using viscous shear theory instead of the normally used Coulomb friction theory by almost all the researchers in this field. In this paper they revealed the potential of using a viscous friction model to model slip in the Van Doorne metal pushing V-belt CVT which according to them predicts the measured slip with good accuracy. They compared their results with that of Becker H. J. 1987 which used a Coulomb friction model. However, the paper did not explained how they calculated the belt slip. Nevertheless, the author would say this initial effort shows interesting results.

The paper by Karam A. and Play D. 1992, was similar in nature to that of Carvajal J. M. et al 1987, reviewing (also assuming Coulomb friction) the metal pushing V-belt CVT as a highly discretised mechanism with a large number of parts, and recognised the difficulties of modelling the mechanical behaviour due to the relative motion of the belt elements.

Fujii T. et al's 1993a and 1993b have done very good studies on the metal pushing V-belt CVT which provide useful information on the belt mechanical behaviour. The 1993b paper studies on the distribution of the block compression force (reproduced here as Fig. 2.4, 2.5 and 2.6) has answered and clarified to a certain extent some of the questions and assumptions made by previous researchers in this field. That is, only segments on the slack side of the bands carry the pushing force while on the return tight side, the segments pushing force is virtually zero. On the driving pulley, the segments pushing force exists only near the pulley exit while on the driven pulley, the segments pushing force exist on a greater arc of the pulley with bigger value than the primary pulley during entry and then begin to diminish quite gradually just before the exit. However, in Fujii T. et al's studies, some of the equations used to justify their results were quoted without any justification. For example their equation (1) in Fujii T. et al 1993a,

$$Q_{DN} = \frac{T_{in}^* \cdot \cos \theta}{2 \cdot \mu \cdot R_{DR}} \quad (2.2.5)$$

which they claimed represent the relationship between thrust ratio and torque ratio, and equation (1), Fujii T. et al 1993b,

$$T = \frac{P \cdot R}{t} \quad (2.2.6)$$

which they claimed to represent the band tension, would probably needs some clarification. On top of these, the term maximum transmittable torque T_{in}^* , was not clearly defined and is quite confusing. Also, their experiments were carried out at quite a low input speed (150 ~ 300 rpm) to really represent the practical belt applications range.

Using a similar concept, Kanehara S. et al 1994 extended the work of Fujii T. et al 1993a and 1993b on the study of metal pushing V-belt type CVT to involve all the six possible forces that are acting on the belt segments around the pulleys. The six forces are listed below and shown diagrammatically in Fig. 2.7.

- i. block compression force E ,
- ii. normal force P (between bands and segment shoulders),
- iii. normal force N (between segment sides and pulley cone surfaces),
- iv. transmitting force F_T (tangential friction force between segment sides and pulley cone surfaces),
- v. radial friction force F_R (radial friction force between segment sides and pulley cone surfaces) and
- vi. ring traction force F_{ST} (tangential friction force between band sides and segment shoulders).

To the author, this was an outstanding piece of work because they are able to measure all the above mentioned forces directly, using strain gauges and a telemeter system with the aid of their specially designed segments (where in Fujii T. et al 1993b, only managed to measure the segments compression force with their specially designed segment).

Kitagawa T. et al 1995, extended the work of Kanehara S. et al 1994 further and included speed ratio change as well. Unfortunately, as in the work of Fujii T. et al's 1993a and 1993b and Kanehara S. et al 1994, the experiments of Kitagawa T. et al were also being carried out at quite a low input speed (150 ~ 300 *rpm*) and again do not represent the practical belt applications range.

Among the most appropriate and extensive work in trying to model the Van Doorne metal pushing V-belt CVT behaviour, the author would say, is the one produced by Micklem J. D. et al 1994a and 1994b. From their studies they extended their viscous shear (Micklem J. D. et al 1991) to elastohydrodynamic lubrication at all the belt interfaces to model the metal pushing V-belt CVT design performance; while most of the earlier generation of researchers assumed Coulomb friction. In the 1994a paper,

Micklem J. D. et al investigated belt torque loss of the metal pushing V-belt CVT experimentally and explained its losses with a proposed semi-empirical model; while in the 1994b paper they applied viscous shear and elastohydrodynamic theory to model the belt performance. These two papers interest the author and quite a number of their concept and ideas were used in this work to further enhance the models that are developed in this thesis.

Another paper published on the Van Doorne metal pushing V-belt CVT mechanism is the one by Asayama H. et al 1995. Asayama H. et al also assumed Coulomb friction in their theoretical work and claimed that the theoretical formulae they derived for calculating tension were in good agreement with their experimental results. In this paper, Asayama main assumptions were that,

- i. the belt is firmly engaged without slip at a certain part of the arc of contact on the larger pulley, which has a larger wrap angle than the smaller pulley, this is because (according to them) equation,

$$\frac{T_2}{T_1} = e^{-\mu_b \cdot \delta} \quad (2.2.7)$$

holds along the arc of contact under varying band tension,

- ii. band tension varies along the arc of contact on the smaller pulley and on the larger pulley the band tension is constant at a certain part of the arc of contact, which they considered as a no-slip zone.

However in section 3.3.1 of their paper, it was not mentioned how β_1 and β_2 (angles of contact under constant compressive force for the driving and driven pulley respectively) were determined (reproduced here as Figures 2.8, 2.9 and 2.10). Also in section 4.1, there were no explanations on how the free body diagram, reproduced here as Fig. 2.11 was achieved and the author feels that it was not quite right. That is, the small increment in band tension dT , should be on the left-hand-side rather than on the right. This is because, the friction force $\mu_b \cdot dP$ was assumed to be acting to the right, thus this will cause the band tension on the left-hand-side to increase by the small amount dT . This in turn introduces an error in his corresponding free body diagram for the segment reproduced here as Fig. 2.12. That is, $\mu_b \cdot dP$ should

now be acting to the right while the one in Fig. 2.11 should be acting to the left. Then in their band tension equation (4), the author feels that the term $\nabla \cdot d\theta$ was left out. Also, an important point to note on Asayama et al's work is that, their results were obtained at a very low input speed ($10 \sim 20 \text{ rpm}$) which probably explained why they neglected the bands centrifugal force component $\nabla \cdot d\theta$ in their equation (4). Thus, this will mean that their theoretical formulae are not suitable for real life applications. However, the bands centrifugal force component seems to appear back in their segments compressive and clamping forces of their equations (7) and (8) respectively, which probably requires an explanation.

Recently, Micklem J. D. et al 1996, published another paper on the Van Doorne metal pushing V-belt CVT, which the author would say summarised most of the work that they have published previously. In this paper, Micklem J. D. et al highlighted that the failure of the metal pushing V-belt CVT to achieve improvement in fuel economy was mainly due to belt torque losses, hydraulic pump loss and loss due to slip in the belt pulley system. Despite these losses, the engine can be controlled to operate near to its region of maximum efficiency. All these losses according to Micklem J. D. et al, could be minimised by the proper adjustment of the secondary pulley clamping pressure. However, they did not reveal the range of its values.

Probably the most recent publications on the Van Doorne metal pushing V-belt CVT are those published in the proceedings of the International Seminar on Advanced Vehicle Transmission and Powertrain Management organised by the IMechE, London from Thursday 25th until Friday 26th September 1997 which, the author fortunately had the opportunity to attend. However, most of the papers published in these proceedings only present and discuss the potentials and the abilities of the Van Doorne metal pushing V-belt CVT. The author would say, none of the papers really discusses the engineering and design of the belt itself. Nevertheless, one of the papers that interested the author most was the one presented by Rine Pelders from Van Doorne Transmisie on the "High Torque Push belt CVT Design and Test Results". In

this paper, he reveals the design and test results of the newly developed P930 transmission. They claimed that the Transmission can now deliver up to 325 *N.m* of torque. This was achieved with the newly developed 2 stage roller pump and the 30 *mm* push belt. He also reveals that in partial load conditions at low and moderate temperature, the transmission efficiency is strongly dependent on the transmission oil temperature.

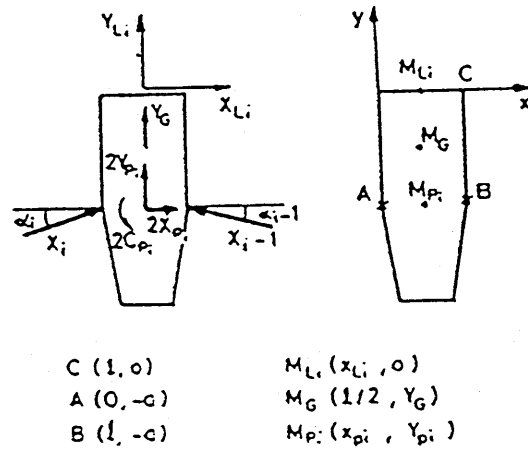


Figure 2.1 Segment equilibrium [Carvajal J. M. 1987].

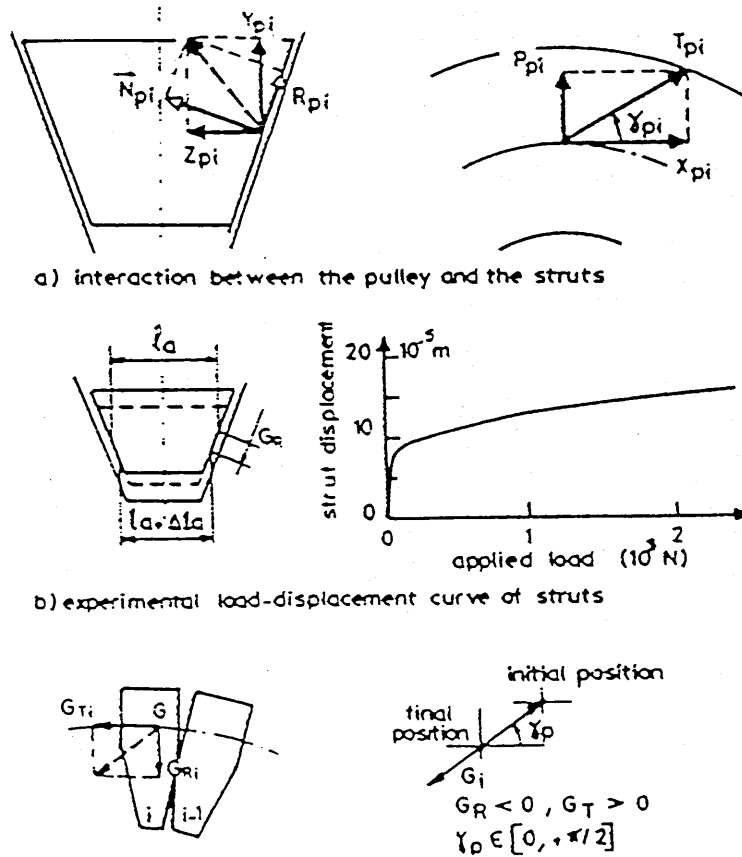


Figure 2.2 Segment equilibrium [Carvajal J. M. 1987].

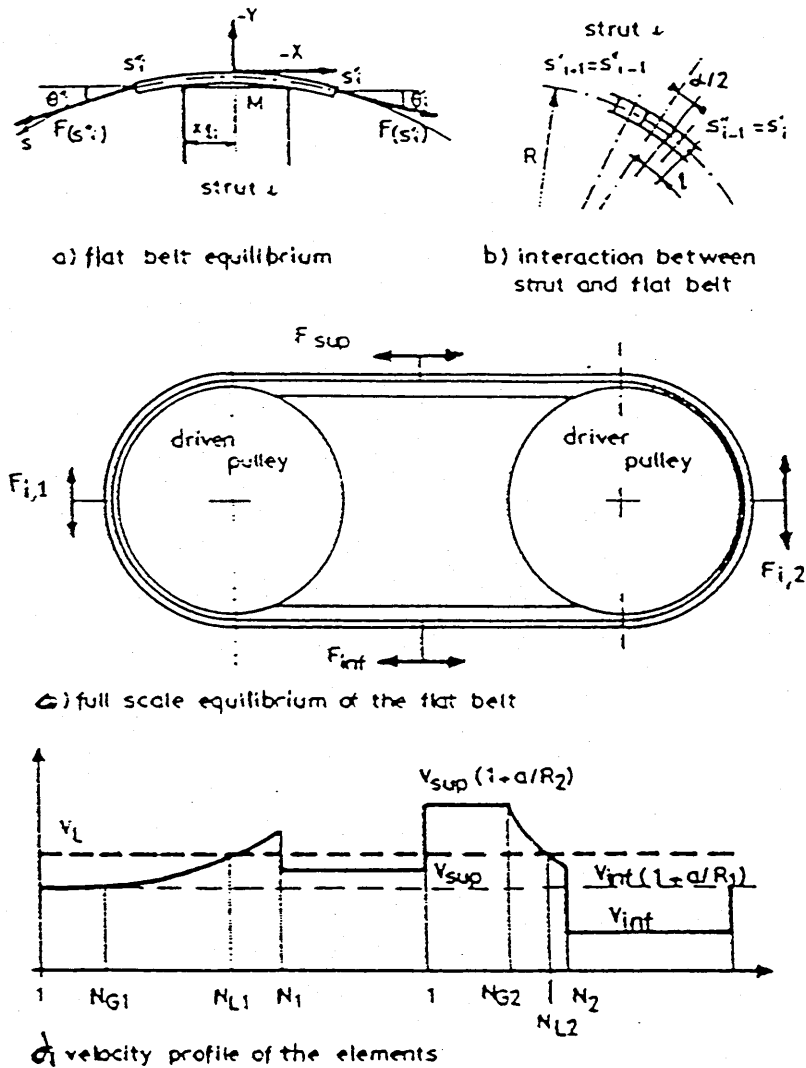


Figure 2.3 Equilibrium of the flat belt and the segment [Carvajal J. M. 1987].

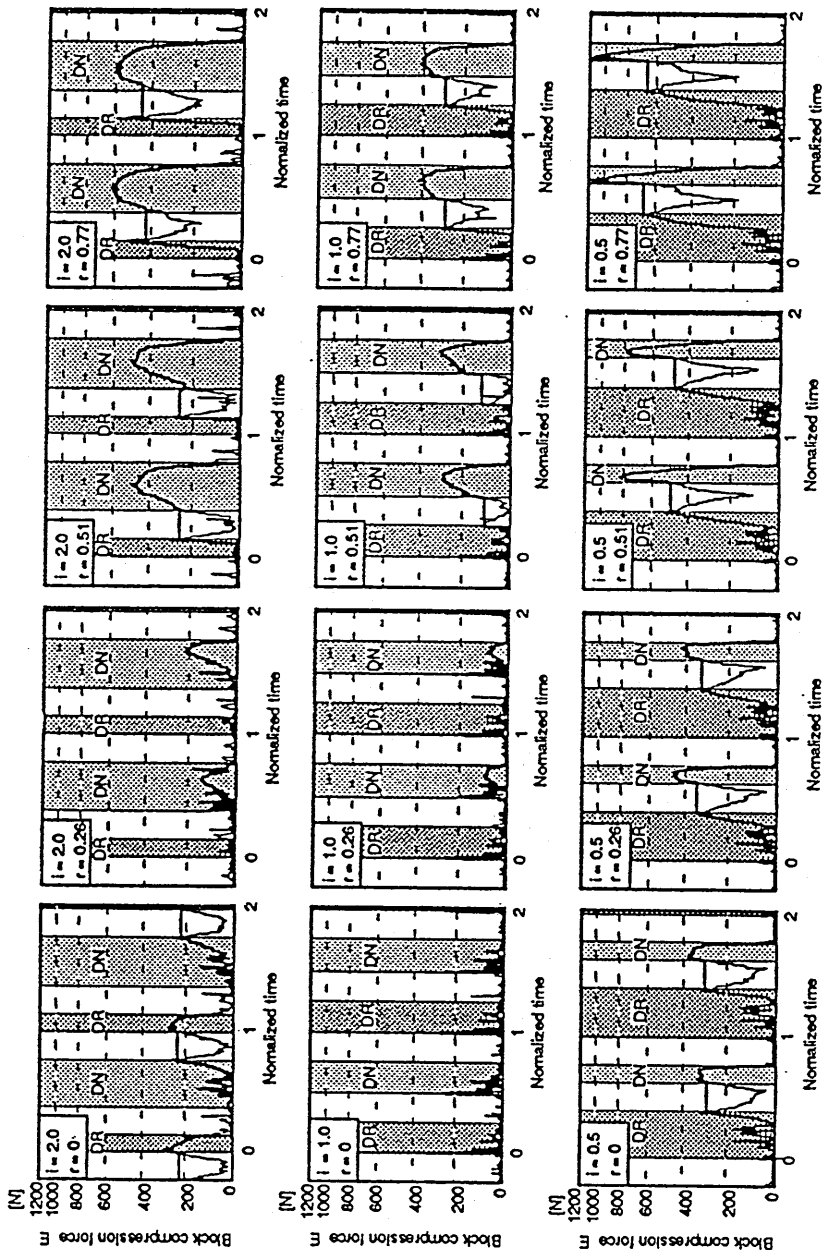


Figure 2.4 Distribution of segment compression force [Fujii T. et al 1993].

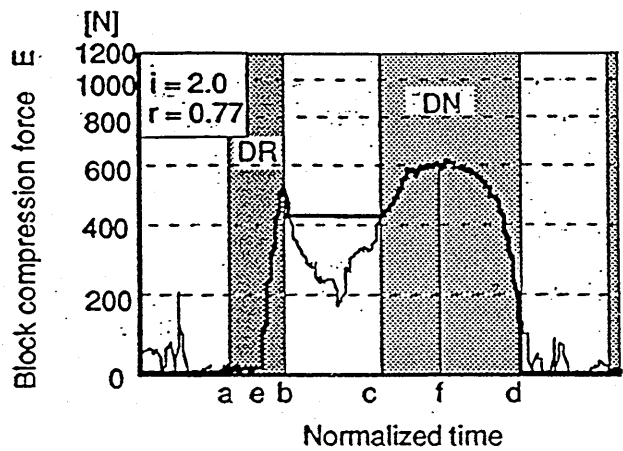


Figure 2.5 Magnified distribution of segment compression force [Fujii T. et al 1993].

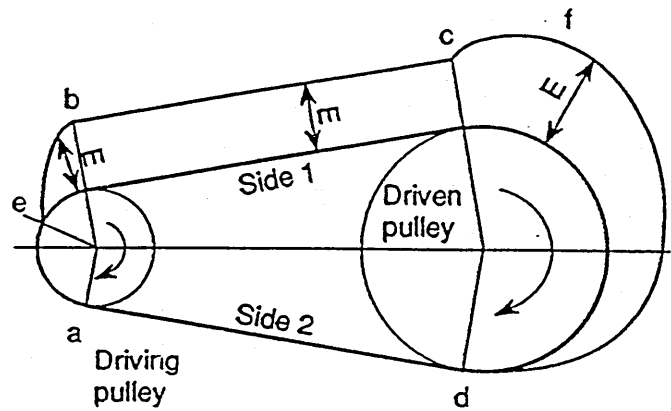


Figure 2.6 Graphic distribution of segment compression force [Fujii T. et al 1993].

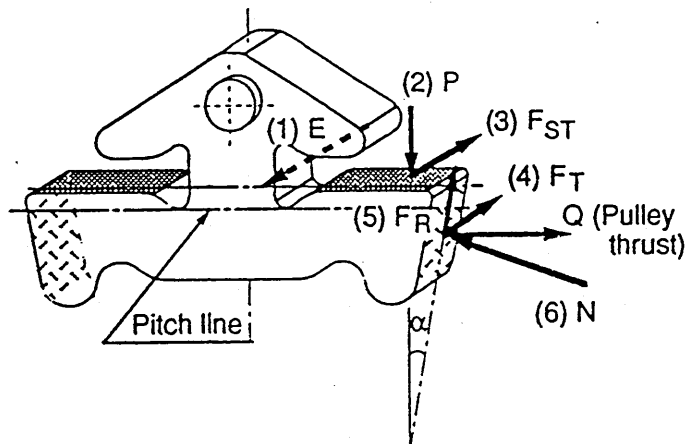


Figure 2.7 Forces acting on a block [Kanehara S. et al 1995].

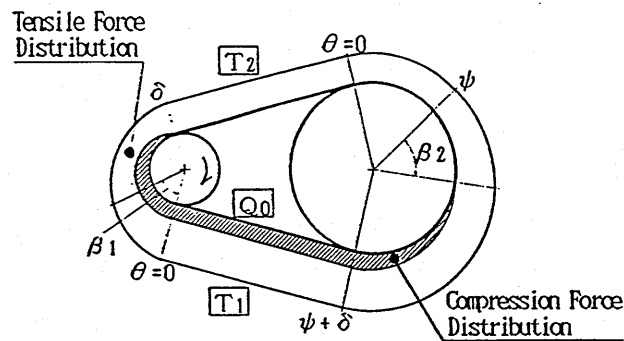


Figure 2.8 Torque transmitting model of pushing type belt (low ratio, small torque) [Asayama H. et al 1995].

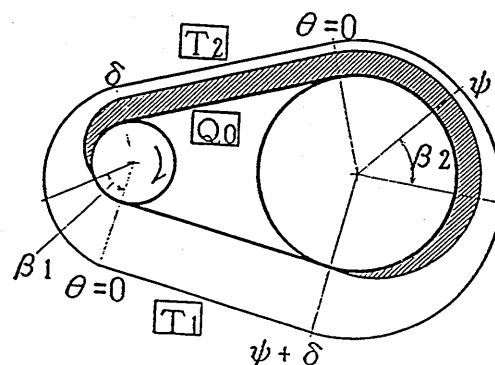


Figure 2.9 Torque transmitting model of pushing type belt (low ratio, large torque) [Asayama H. et al 1995].

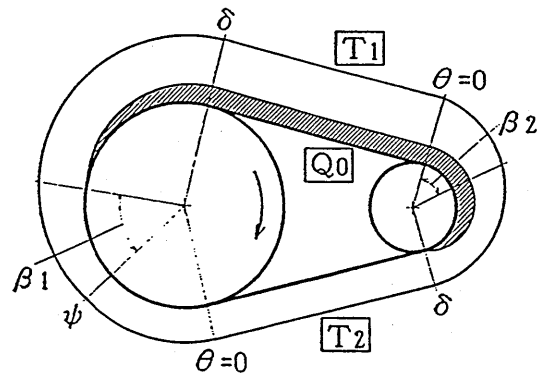


Figure 2.10 Torque transmitting model of pushing type belt (overdrive) [Asayama H. et al 1995].

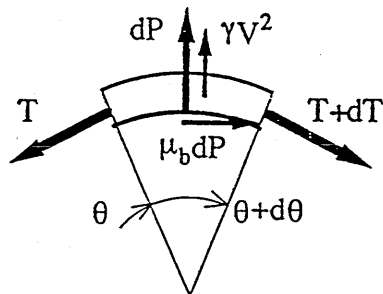


Figure 2.11 Forces acting on the band [Asayama H. et al 1995]

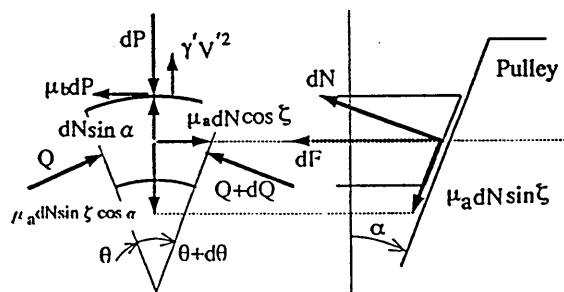


Figure 2.12 Forces acting on the block [Asayama H. et al 1995]

Belt/pulley interface (all values for suitable centre distance)				
	24 mm width		30 mm width	
number of rings	belt type	$T_{\max}/n_{\max}$	belt type	$T_{\max}/n_{\max}$
6	24/6	80/7500		
7				
8			30/8	200/6500
9	24/9	125/6500		
10	24/10	140/6500	30/10	250/6500
11				
12	24/12	165/6500	30/12	300/6500

Table 2.1 Belt Differentiation Scheme [Roovers G. 1991]

CHAPTER 3

MISALIGNMENT EQUATION FOR THE VAN DOORNE METAL PUSHING V-BELT CVT

3.1 Introduction

In the world of continuously variable transmissions (CVTs) for automotive applications, the Van Doorne metal pushing V-belt CVT has been one of the most successful. It has managed to overcome most of the problems and limitations for it to be mass produced. Ford Fiesta CTX, Subaru Justy and Nissan Micra are a few cars that have been mass produced and these cars are able to compete with the manual gearbox transmission automobiles of the same ranges. For example, the Nissan Micra 1.3L CVT has been rated "The Perfect City Car" by the Road Test Year Book 1993 (published by Autocar Motor magazine). All these indicate that limitations like cost, durability, efficiency, packaging and reliability have all reached the necessary standards for automotive applications.

The existence of belt misalignment in the Van Doorne metal pushing V-belt continuously variable transmission (CVT) has always been recognised by researchers and the manufacturers themselves. However until now, the effect of misalignment faced by the Van Doorne metal pushing V-belt CVT has not been investigated. Its effect has always been assumed to be negligible and search through the literature reveals no work published so far to verify this assumption. Sun D. C. and Watson T. J. 1988, recognised the existence of misalignment but neglected it. Hendriks E. et al 1988, estimated the magnitude of the misalignment by an equation and were able to minimise it in the design, also ignored the misalignment effect on the transmission performance. Micklem J. D. 1990, produced an outstanding viscous shear stress theory in predicting the transmission performance and Micklem J. D. et al 1994a and 1994b, in continuing the effort of improving Micklem J. D. 1990's work into viscous shear and elastohydrodynamic lubrication model, also ignored the misalignment effect.

To the best of the author's knowledge, there was only one published paper that discussed the misalignment of the Van Doorne metal pushing V-belt CVT, that is by Hendriks E. et al 1988. In that paper, the misalignment of the Van Doorne metal pushing V-belt CVT was estimated by the following equation,

$$\Delta = \frac{(2 \cdot RB_{med})^2}{\pi \cdot X} \cdot \left(\frac{i-1}{i+1} \right)^2 \cdot \tan \theta \quad (3.1.1)$$

However, this misalignment equation only approximately estimates the misalignment of the belt, if it was initially assembled to have zero belt misalignment (i.e. the ideal straight plane) at speed ratio 1:1.

Thus, the lack of information on the belt misalignment values and its geometrical characteristics, hopefully would justify the author's work in this area. It is felt that these misalignment values will be important to future designers, especially those involved with the belt control system. It could also provide important information for further improvement of the Van Doorne metal pushing V-belt CVT design. Also, misalignment of the belt may increase losses. This is because, belt misalignment will cause friction and energy losses due to additional sliding of the belt segments between themselves and the pulleys cones surfaces, which in turn will reduce the Van Doorne metal pushing V-belt CVT performance and efficiency. This initial effort by the author is to address these areas and study the geometrical behaviour of the belt misalignment theoretically. An equation to represent the belt geometrical behaviour has been developed and it is able to calculate the magnitude and position of the belt misalignment at different speed ratios and at different assembled zero belt misalignment speed ratios. The condition necessary to minimise the magnitude of the misalignment has been determined and verification of the equation, was obtained by using a solid modeller from a Unigraphics Version 10.4.0 CAD package. The belt misalignment equation developed from this chapter will then be used to further investigate and analyse the effect of misalignment on the efficiency and performance of the transmission (Chapter 5).

3.2 Derivation Of The Belt Misalignment Equation

Geometrically, the Van Doorne metal pushing V-belt CVT will behave exactly like any ordinary rubber V-belt. Thus, classical V-belt theory can be applied to it. For the derivation of this particular misalignment equation, it is assumed that the belt behaves ideally. That is,

- i. the belt can be represented by an undeformable inextensible flexible material
- ii. the belt will enter and exit the pulleys tangentially
- iii. the belt is uniform and continuous and
- iv. there is no power loss.

From classical V-belt theory (Fig. 3.1),

$$L = RB_p \cdot \beta_p + RB_s \cdot \beta_s + 2 \cdot \sqrt{X^2 - (RB_s - RB_p)^2} \quad (3.2.1)$$

$$\beta_p = \pi - 2 \cdot \alpha \quad (3.2.2)$$

$$\beta_s = \pi + 2 \cdot \alpha \quad (3.2.3)$$

and,

$$\sin \alpha = \frac{(RB_s - RB_p)}{X} \quad (3.2.4)$$

Rearranging equations (3.2.1) to (3.2.4) we get,

$$RB_p = \frac{[L - X \cdot \beta_s \cdot \sin \alpha - 2 \cdot X \cdot \sqrt{(1 - \sin^2 \alpha)}]}{2 \cdot \pi} \quad (3.2.5)$$

and,

$$RB_s = X \cdot \sin \alpha + RB_p \quad (3.2.6)$$

By varying the angle α , correspondingly the values of β_p and β_s can be tabulated. Consequently from these values, the appropriate values of RB_p and RB_s can be calculated if we know the belt centre distance X and length L (which are usually dictated by the transmission designer). From here, a series of drawings for various speed ratios can be drawn to give a clear picture of the belt misalignment geometrical behaviour.

Let's first assume that the belt is initially assembled so that the ideal straight plane or zero belt misalignment (i.e. $\Delta = 0$) will take place at speed ratio 1:1. For this

particular position, let the primary and secondary pulleys radii be RB_{mP} and RB_{mS} respectively, as shown in Fig. 3.1. The figure shows the overdrive, 1:1 and underdrive position being superimposed over each other to illustrate the belt and half-pulley movements during speed ratio change. Let RB_{P1} and RB_{P2} be the appropriate primary pulley radii, and correspondingly RB_{S1} and RB_{S2} be the secondary pulley radii obtained from equations (3.2.5) and (3.2.6). During speed ratio change, it can clearly be seen that the belt slides axially along the pulleys cones surfaces (at an angle θ) of the fixed halves of both pulleys, while the adjustable halves ensure the exact speed ratio position and keep the belt taut during application. If the mid-points of the belt on the primary (P_m , P_{m1} and P_{m2}) and secondary (S_m , S_{m1} and S_{m2}) pulleys were to be plotted during speed ratio change, their locus will describe a straight line passing through the initially assembled ideal straight plane at speed ratio 1:1 (as shown in Fig. 3.2). The line PP represents the locus motion of the belt mid-points at the primary pulley and line SS at the secondary pulley, while the x-axis represents the initially assembled ideal straight plane.

Now, points P_m and S_m lie exactly on the x-axis or the initially assembled ideal straight plane at zero belt misalignment. As the speed ratio increases from 1:1 to an underdrive position, point P_m will slide parallel to the primary pulley cones surfaces along the locus line PP to point P_{m1} . Correspondingly, point S_m on the secondary pulley will slide parallel to the secondary pulley cones surfaces along the locus line SS to point S_{m1} . Similarly, if the speed ratio decreases from 1:1 to an overdrive position, point P_m will slide to point P_{m2} , while point S_m to point S_{m2} . At speed ratio

$i = \frac{RB_{S1}}{RB_{P1}}$, the misalignment of the belt on the primary pulley,

$$\Delta_1 = \Delta_{S1} - \Delta_{P1} \quad (3.2.7)$$

Now, for the primary pulley,

$$\Delta_{P1} = (RB_{mP} - RB_{P1}) \cdot \tan \theta \quad (3.2.8)$$

Similarly, for the secondary pulley,

$$\Delta_{S1} = (RB_{S1} - RB_{mS}) \cdot \tan \theta \quad (3.2.9)$$

Substituting equations (3.2.8) and (3.2.9) into equation (3.2.7), gives misalignment,

$$\Delta_1 = [(RB_{S1} - RB_{mS}) - (RB_{mP} - RB_{P1})] \cdot \tan \theta \quad (3.2.10)$$

where Δ_1 , is the misalignment of the belt at speed ratio equal to $\frac{RB_{S1}}{RB_{P1}}$. Consequently,

Δ_2 will be the misalignment at speed ratio equal to $\frac{RB_{S2}}{RB_{P2}}$ and so on. So, if the belt is

initially assembled in such a way that the ideal straight plane will take place at speed ratio other than 1:1, the locus line PP and the corresponding locus line SS will be offset to their respective new P_m and S_m points. The locus will still be a straight line passing through the new P_m and S_m points at the same angle θ , to the x-axis. Hence, equation (3.2.10) will still apply for calculating the misalignment values. Therefore, the misalignment equation for the belt can then be generalised as,

$$\Delta = [(RB_S - RB_{mS}) - (RB_{mP} - RB_P)] \cdot \tan \theta \quad (3.2.11)$$

By equating Δ to zero from equation (3.2.11) we can then locate the lateral displacement of the second ideal straight plane position. That is,

$$(RB_S - RB_{mS}) = (RB_{mP} - RB_P) \quad (3.2.12)$$

This indicates that the ideal straight plane/s or zero misalignment/s will take place when the left-hand-side of equation (3.2.12) equals to the right-hand-side of the equation. Hence, as long as both sides of the equation are equal, ideal straight plane conditions exist. If the system was initially assembled so that the ideal straight plane position will take place at speed ratio, $i = 1.0$ (i.e. RB_{mS} , RB_{mP} , RB_S and RB_P are equal during assembly), there will only be one ideal straight plane position; but, if the system were initially assembled so that the ideal straight plane position would take place at speed ratio other than 1.0 (i.e. $RB_{mS} \neq RB_{mP}$ during assembly), then there would always be two ideal straight plane positions as has been highlighted graphically by Hendriks E. et al 1988. The lateral displacement e (Fig. 3.3 and 3.4), of the second ideal straight plane position would then be given by the value of the left-hand-side or right-hand-side of equation (3.2.12) multiplied by $\tan \theta$.

3.3 Sample Calculations

From Micklem J. D. 1990, manual drawings were constructed (and then justified by computer aided solid model using Unigraphics Solid Modeller Ver.10.4.0), then a series of misalignment values were obtained from the Unigraphics model and calculated using equation (3.2.11). These were found to agree and are shown in Fig. 3.3 and 3.4 for overdrive and low speed ratio positions of 0.700046 and 1.28770 respectively (please note that the drawings were not drawn to scale). These values were then compared with the calculated values using equation (3.2.11) as plotted in Fig. 3.5. These values agree exactly, thus justifying the accuracy of the developed belt misalignment equation. The calculated values also confirmed that the second zero belt misalignment position takes place at exactly the inverse of the initially assembled speed ratio of the ideal straight plane. An example of calculating the lateral displacement e , is given below.

$$i = \frac{44.08578}{62.97554} = 0.700046$$

where,

$$RB_{mP} = 62.97554 \text{ mm and } RB_{mS} = 44.08578 \text{ mm}$$

i.e. the ideal straight plane will initially take place at

$$RB_{P1} = 62.97554 \text{ mm and } RB_{S1} = 44.08578 \text{ mm}$$

while the second ideal straight plane position will occur at

$$RB_{P2} = 44.08578 \text{ mm and } RB_{S2} = 62.97554 \text{ mm}$$

Therefore, the lateral displacement e , of the second ideal straight plane position at speed ratio, $i = 0.700046$ will be given by

$$\begin{aligned} e_{i=0.700046} &= (RB_{mP} - RB_{P2}) \cdot \tan \theta \\ &= (RB_{S2} - RB_{mS}) \cdot \tan \theta \\ &= (62.97554 - 44.08578) \cdot \tan 11^\circ \\ &= 3.67179 \text{ mm.} \end{aligned}$$

(Note: The sign is positive, indicating that the second ideal straight plane position should be on the positive y-axis, as shown in Fig. 3.3).

3.4 Discussion

The comparison of results obtained from the solid model generated using Unigraphics and those obtained from equation (3.2.11) show complete agreement. This justifies the accuracy of equation (3.2.11). Fig. 3.3 and 3.4 are diagrammatic representations of the belt positions and misalignments based on the condition of no belt misalignment for speed ratios of 0.700046 and 1.28770, while Fig. 3.5 shows a graphical presentation of the misalignment for the same conditions using equation (3.2.11). Manual drawings also showed similar results further verifying the accuracy of equation (3.2.11). However, when using these equations, the user must bear in mind that the values obtained are not an exact representation of the belt misalignment. That is, the friction between all the possible sliding surfaces (as highlighted by all the previously mentioned researchers) that might affect the geometrical behaviour of the belt were ignored. The equations actually represent the ideal belt misalignment behaviour which will serve as an ideal standard for the belt misalignment values.

The equations for calculating the belt misalignment values when the transmission is in running condition were derived with the assumption that the belt behaves ideally. This of course is unlikely to happen practically. The belt will definitely bend gradually and not instantly as shown in Fig. 3.3 and 3.4 due to the curvature of the pulleys cones surfaces. The segment elasticity may also contribute to extra sliding during exit from the pulleys, thus preventing the belt from misaligning immediately. This will then result in sliding friction and introduces additional energy loss. Similar events will also take place as the belt enters the pulleys. Also, as shown by the results in Fig. 3.5, the minimisation of the belt misalignment can be calculated instead of using the approximate graphical method by Hendriks E. et al 1988. By comparing the belt misalignment values from the equation estimated by Hendriks E. et al 1988 with the author's belt misalignment equation, it was found that the former estimation is only accurate for speed ratio range of about 0.8 to 1.2 (as shown in Fig. 3.6). At speed ratio of 0.77658, the difference in misalignment between the results of equation (3.2.11) and Hendriks E. et al 1988's equation is $4.59 \times 10^{-4} \text{ mm}$ (0.630%). Similarly at speed ratio 1.25514, the values are $3.01 \times 10^{-4} \text{ mm}$ (0.509%).

Practically, the application speed ratio range of an automotive transmission is between 0.5 to 2.5. If the Van Doorne metal pushing V-belt CVT is being assembled so that the zero belt misalignment will take place at speed ratio 1:1, the maximum belt misalignment will be approximately 0.8 mm at speed ratio 2.5 (Fig. 3.5). In order to minimise this, the transmission should be assembled with zero belt misalignment when the speed $i \approx 0.54$ (as shown in Fig. 3.5) with a maximum belt misalignment of approximately 0.40 mm. Alternatively, as the transmission operates at a speed ratio $i \approx 0.5$ for the majority of its life, zero belt misalignment can be arranged for this condition. This result in a maximum belt misalignment of approximately 0.49 mm at a speed ratio $i = 1.00$ and with a maximum reversed belt misalignment of approximately 0.31 mm at a speed ratio of 2.5.

3.5 Conclusions

The study has shown that, the Van Doorne metal pushing V-belt CVT misalignment geometrical behaviour can be ideally modelled by mathematical equations. From the author's misalignment equations, the belt misalignment values can be calculated, the gradient sign of the belt with respect to the ideal straight plane is known (with some modification of the equation that includes the ideal lap angles β_p and β_s , the actual magnitude of the gradient can also be calculated) and the second ideal straight plane lateral displacement from the initially assembled ideal straight plane can also be found. The data obtained from these findings, can serve as useful information to further investigate the effect of belt misalignment on the efficiency and performance of the transmission.

3.6 Figures

Figure 3.1 Misalignment illustration of the Van Doorne Metal Pushing V-belt CVT for various speed ratios.

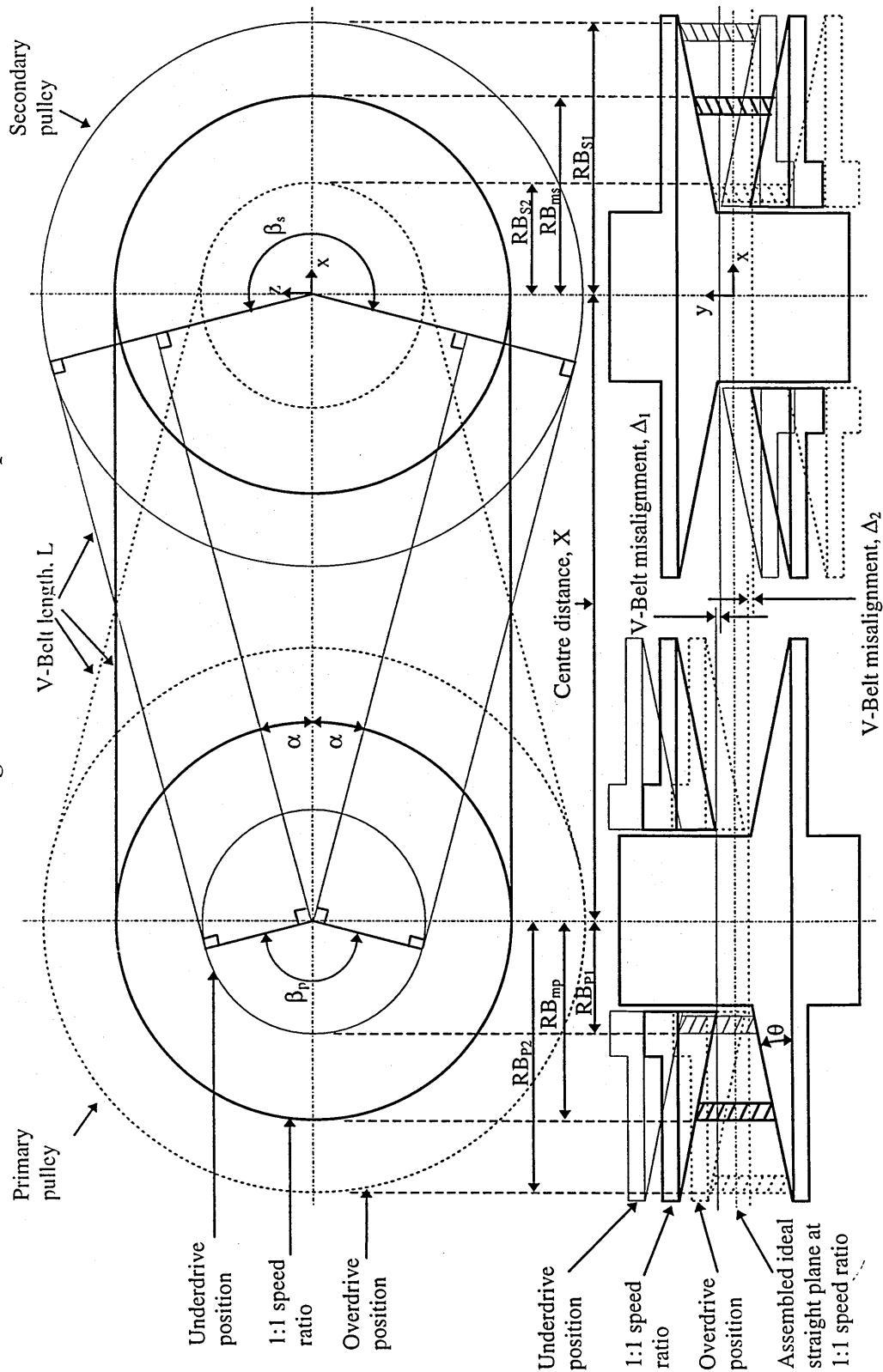
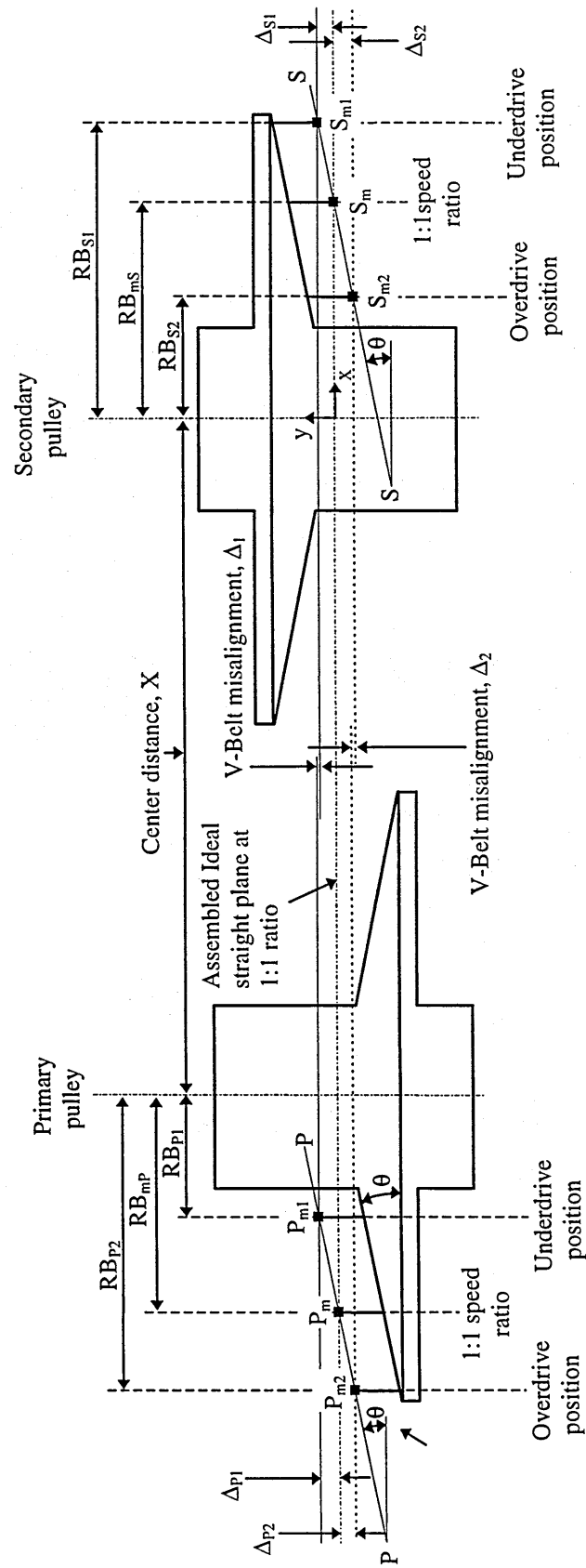
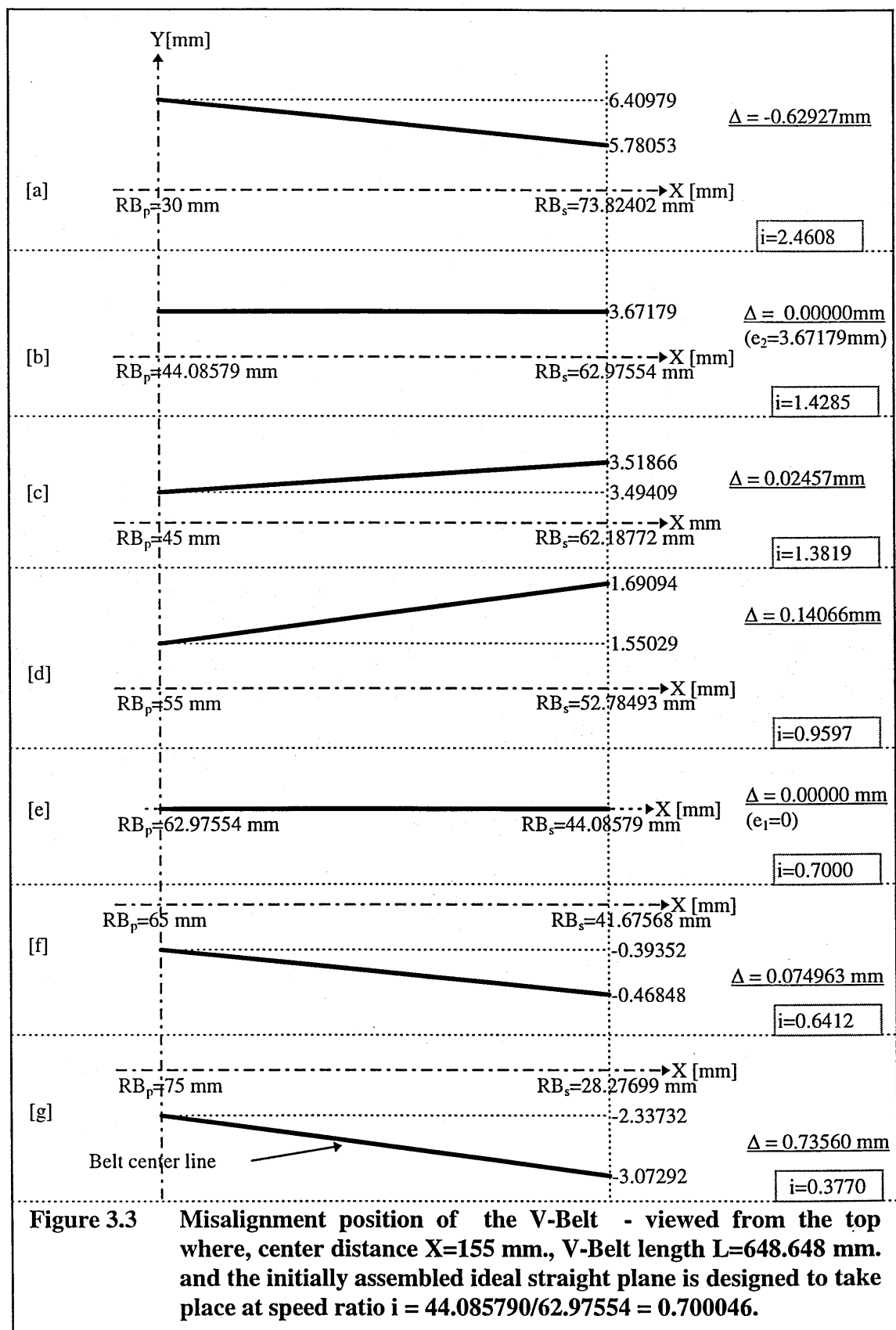


Figure 3.2 Van Doorne Metal Pushing V-belt CVT middle-points locus for various speed ratios.





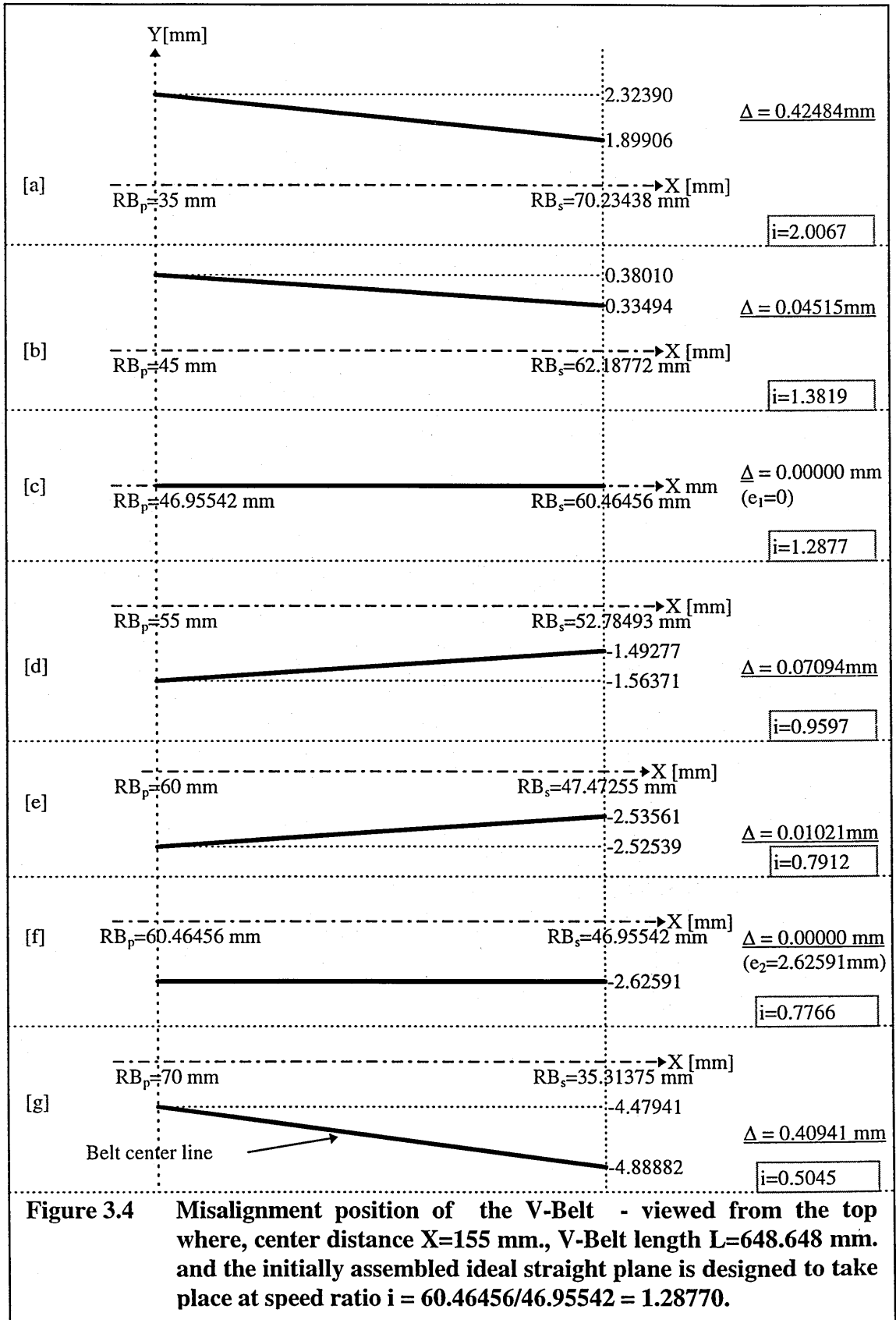


Figure 3.5 Misalignment graphs plotted from the author's equation for various ideal straight plane positions.

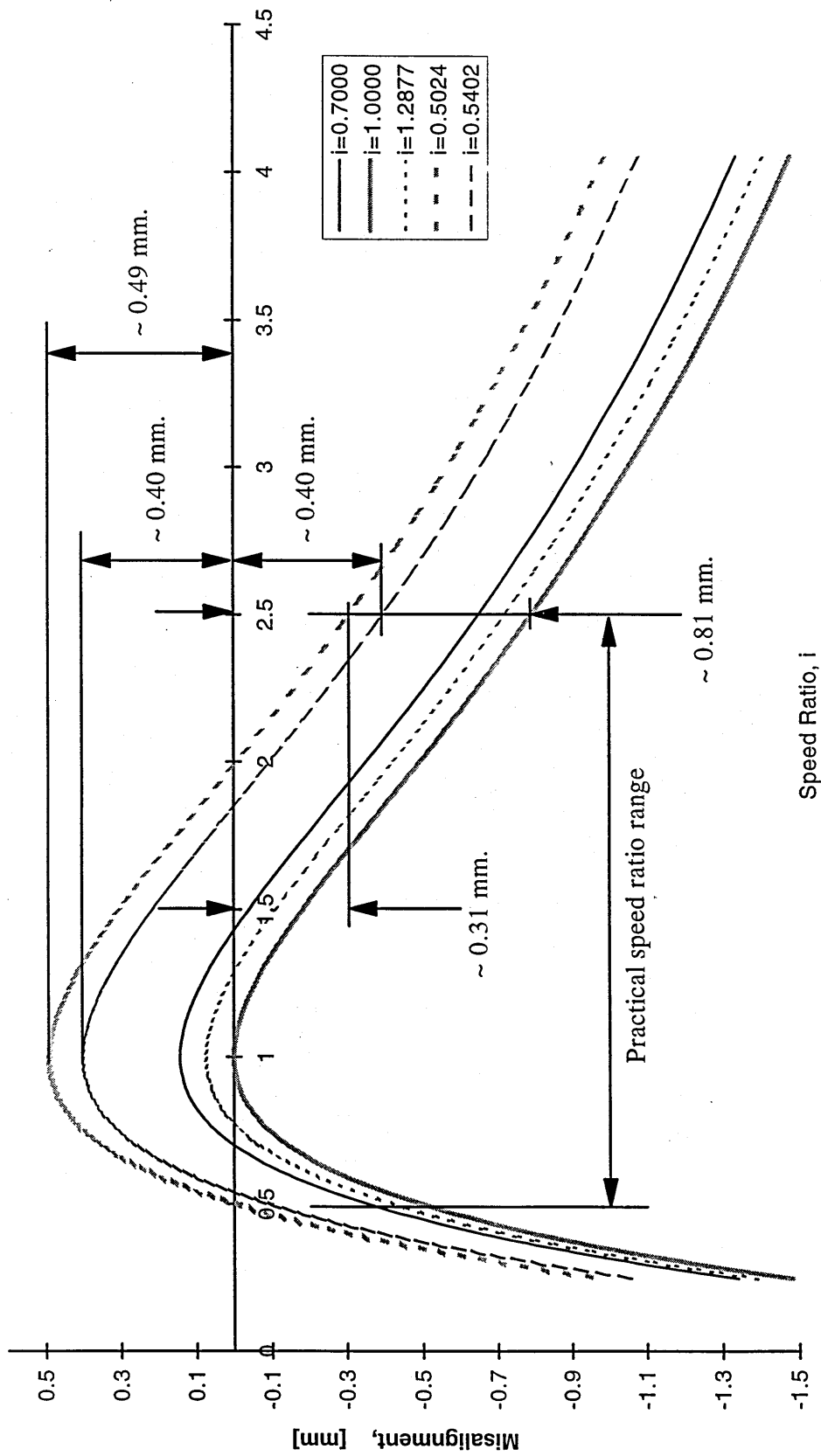
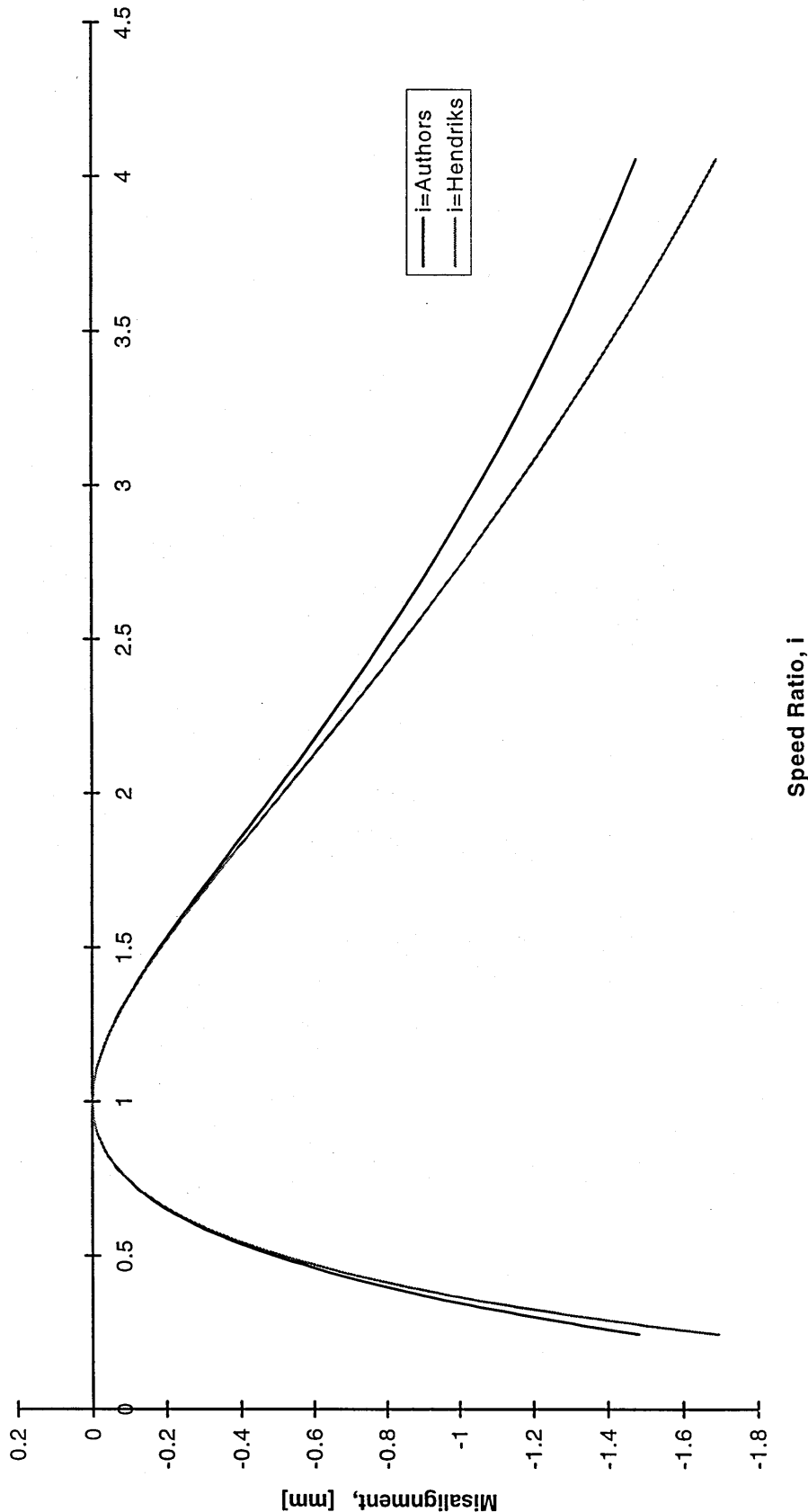


Figure 3.6 Misalignment comparison between Hendrik E et al 1988's estimation (replotted by the author with inverse sign) with author's values.



CHAPTER 4

MISALIGNMENT MODEL OF THE VAN DOORNE METAL PUSHING V-BELT CVT

4.1 Introduction

From the studies made so far by the author, it was found that, to mathematically model the work done or power loss due to belt misalignment of the Van Doorne metal pushing V-belt CVT is a quite a complex problem due to the many parameters involved. This is because, most of the Van Doorne metal pushing V-belt CVT components can move relative to each other which makes it very difficult to model mathematically. This is probably why most of the previous researchers ignored its effect or assumed that its effect is negligible which further justify the author's work in this area. A few models or concepts have been put forward by the author in trying to model the work done and power loss of the Van Doorne metal pushing V-belt CVT due to its belt misalignment. In general, all the models considered assumed that the belt will leave the pulleys cones tangentially.

4.2 First Model

In this first model, the segments were assumed to stay butted together at one of its ends and tend to rotate accordingly to the curvature of the pulleys cones surfaces, as shown in Fig. 4.1 before leaving the pulley freely. Due to the clamping action and the greater radius of curvature during exit, the opposite ends of the segments will have to travel a greater distance and at a greater speed until the segments recover their full free length. Thus in doing so, sliding will take place between these sides of the segments and the pulley cone surface. Hence, friction work done during the sliding motion leads to some power loss.

Considerable development has been made to this initial model. However, finally this model or concept was found to be inappropriate to mathematically model the work done or power losses of the transmission due to the belt misalignment. It failed to explain how the belt misalignment affects the performance and efficiency of the transmission.

This is because, by assuming that the segments will stay butted together at one end and tend to rotate accordingly to the curvature of the pulleys cones surfaces before leaving the pulleys freely, means that:-

- a) As shown in Fig. 4.2, at zero belt misalignment, the segments will be compressed against the pulley cone surface (surface $a_p b_p c_p$), by the other halves of the pulley cone surface (surface $a'_p b'_p c'_p$). Thus, sliding will have to take place along the segments front and back surfaces. This proves that the hypotheses is inappropriate because at zero belt misalignment, there should not be any sliding between the segments themselves.
- b) The segments will have to leave the pulley cone surface tangentially at c_p for a belt misalignment Δ say as shown in Fig 4.3. However, the full scale geometrical studies made (using Unigraphics Solid Modeller, Ver. 10.4.0), show that the radius of curvature of the pulley cone cross-sectional-area curve is always smaller than the radius of curvature of the segments which resulted from the belt misalignment Δ . The segments drawn as dotted lines clearly show that, if the segments were to leave the pulley cones tangentially according to the pulley cone surface radius of curvature, the belt misalignment Δ will have to be huge. Bearing in mind that the magnitude of the belt misalignment Δ , is in the range of $0 \sim 1.00 \text{ mm}$ and not as has been exaggerated in Fig. 4.3 (even then the radius of curvature of the misaligned segments is still less than the pulley cone cross-sectional-area curve). This, clearly indicates that the assumption made is wrong. Thus, the segments will then have to leave the primary pulley cones earlier before reaching c_p . Also, as shown in Fig. 4.3, the segments will have to rotate in one direction and then in the opposite.
- c) Also, this model assumes that the segments do not deformed equally due to the pulley compression force but the deformation will only occur at one end of the segments as shown in Fig. 4.2 and 4.3. This also is not correct.

Hence, this model or concept was finally abandoned.

4.3 Second Model

After realising that the first model was inappropriate, different assumptions were made regarding the behaviour of the belt misalignment when it leaves and enters the pulleys. This time, it was assumed that the segments were compressed equally at both ends and will only rotate (due to belt misalignment) once it has been fully released from the pulley clamping force. That is, once the segments have recovered their full free length, the segments will tend to form an 'S' shape curve along the compression straight as shown in Fig. 4.4. By assuming this geometrical behaviour, almost all the necessary geometrical details of the belt can be mathematically calculated.

However, it was also found that this second model or concept failed to explain how the belt misalignment affects the transmission performance and efficiency. This model indicates that, belt misalignment has very little or no effect at all. This is because, by assuming this geometrical behaviour, there is no sliding between the segments and the pulleys and also no sliding between the segments themselves. The only sliding that will occur will be between the segments and the bands, where the losses are relatively very small because:-

- a) The force between them should be relatively very small since the segments and the bands were assumed to stay relatively plain along the compression straight (section 5.2.1). Thus, relatively negligible force will be acting between them to produce any friction force that will lead to power loss when sliding due to belt misalignment.
- b) The compression force which develops along the compression straight will tend to compress the segments and make them behave like a column made up of short struts. Due to this, the segments will not be able to behave as it has been assumed in this model. Thus, no extra work is required to rotate the segments and hence no power loss. Therefore, the transmission efficiency and performance will not be affected due to the belt misalignment.

These therefore, contradict the author's hypotheses that belt misalignment does affect the efficiency and performance of the transmission. Hence, the segments will have to

slide sideways due to the belt misalignment Δ , which finally led the author to the third concept and the proposed model for this work.

4.4 Third and The Proposed Belt Misalignment Model

The Van Doorne metal pushing V-belt CVT segments were designed in such a way that the segments are kept together via a pin and hole, located at the front and back of each segment respectively. The slots at both sides of each segment allow them to slide along the two sets of tension bands. Around the pulley, the segments will be clamped equally together by the pulley cones which restricted their sideways movements while the two sets of tension bands prevent them from flying radially outwards. Along the two straights, the segments are being guided only by the two sets of tension bands, which fit nicely in the segments slots and serve like a pair of rails for the segments to slide forward or backward and also preventing them from flying off. The two sets of tension bands are themselves guided by the pulley cone surfaces and are kept apart by the segment central columns.

Along the compression straight, as shown shaded in Fig. 4.5, the segments are as if being clamped together by a pair of segments, one at the immediate exit of the primary pulley and the second is the one just before entry to the secondary pulley. These two segments resemble the two 'flat plates' between the two jaws of a bench vice. Thus, when belt misalignment takes place, the segments between these two 'flat plates' along the compression straight will tend to slide transversely (or sideways) along their contact surfaces. One might say that, this is not possible because the segments are being held together by the segments pins and holes system. However, the tolerance between the pins and holes allows these movements to take place.

For example, the minimum number of segments compressed or clamped between the two pulleys in this work, $n \approx 66$ (maximum is approximately equal to 77), and the tolerance between the pin and hole, $h \approx 0.05 \text{ mm}$. Then, the total possible belt misalignment Δ that could take place will be equal to $(n-1) \cdot h \approx (66-1) \cdot 0.05 \approx 3.25 \text{ mm}$. This is quite a large belt misalignment movement, and if it is allowed to take place, it is assumed that the amount of power

loss to overcome just the sliding friction between the segments will be significant. This is because, the segments themselves are at the same time travelling at a certain velocity. However, if the belt misalignment goes beyond this possible belt misalignment value, the segment pins and holes will be subjected to side forces. Thus, further losses will result not due to segments sliding between them anymore, but due to segment pin and hole deformations, vibration etc. which will not be covered by this work.

Even if the tolerance between the pins and the holes is to be reduced by 50% to 0.025 *mm*. (which will of course incur an increase in the cost of production), the maximum possible belt misalignment will be reduced to 1.625 *mm*. This relatively, is still a large belt misalignment movement. However, the pin and hole on each segment are designed with the main objective of guiding the segments so as to enable them to align themselves during operations. Thus, a small tolerance could result in the segments failing to perform the required task. Also, this could increase the friction forces between the segments pins and the holes, which will lead to greater power loss, especially around the pulleys. Thus the tolerance between the segments pins and holes is an essential requirement and the segments should be lubricated, and it should also be designed in such a way that it is just enough for the above to take place and not too big to cause excessive vibration.

4.5 Figures

Fig. 4.1 First Belt Misalignment Model

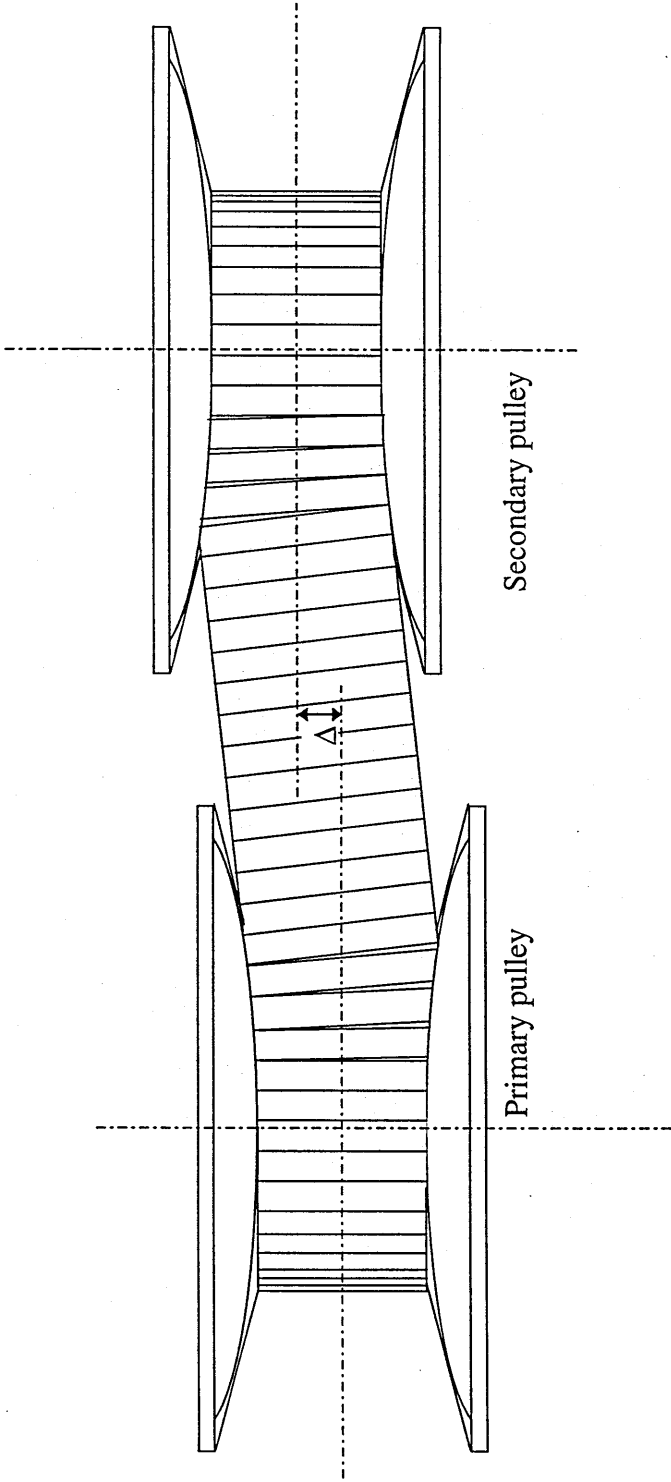


Fig. 4.2 First Belt Misalignment Model
- some details of primary pulley

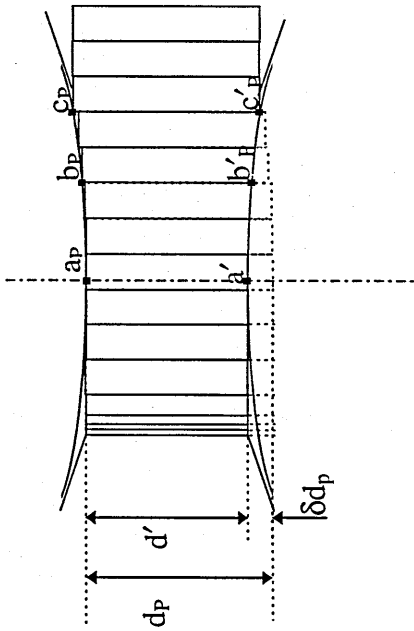


Fig. 4.3 First Belt Misalignment Model - some details of both primary and secondary pulleys

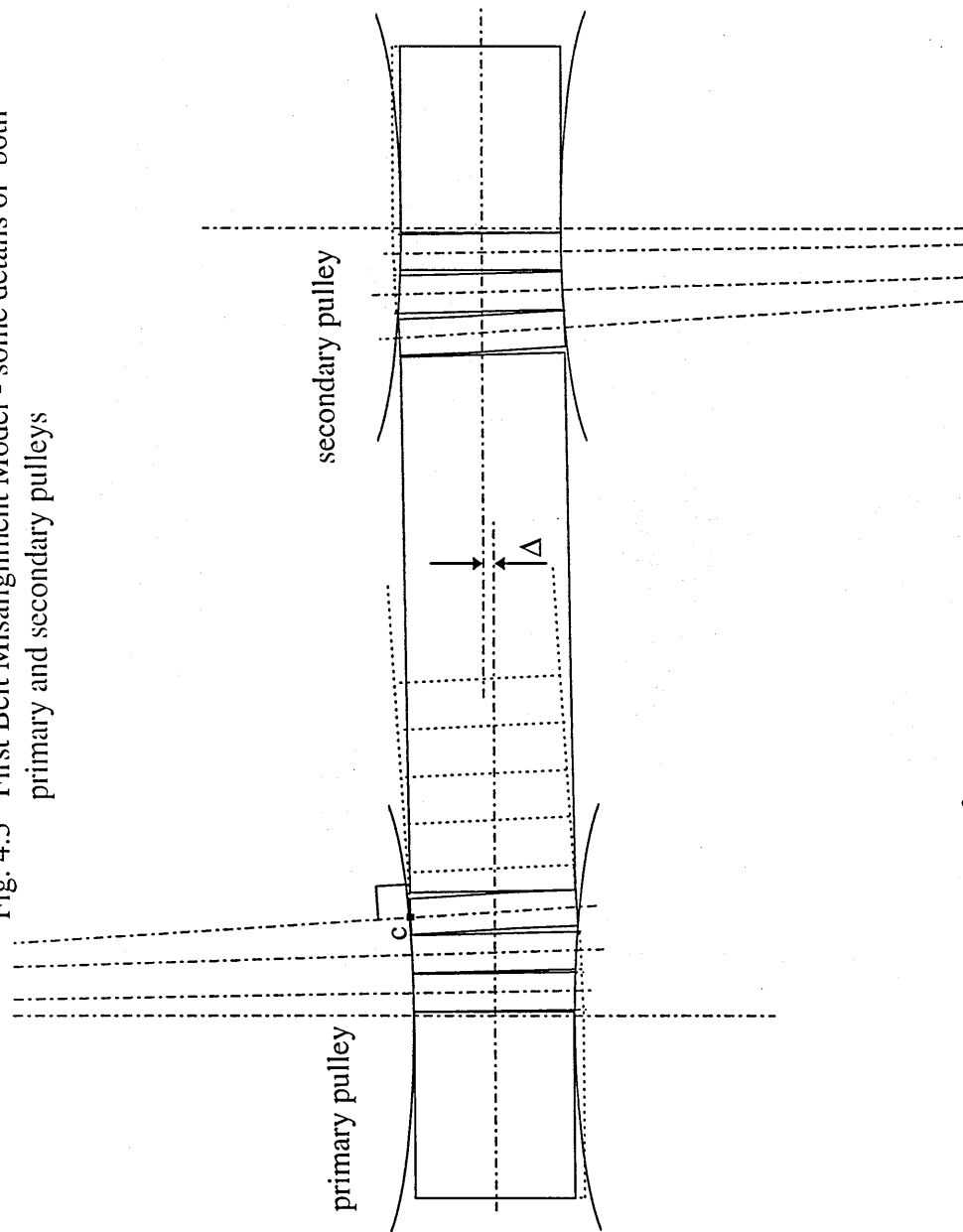


Fig. 4.4 Second Belt Misalignment Model

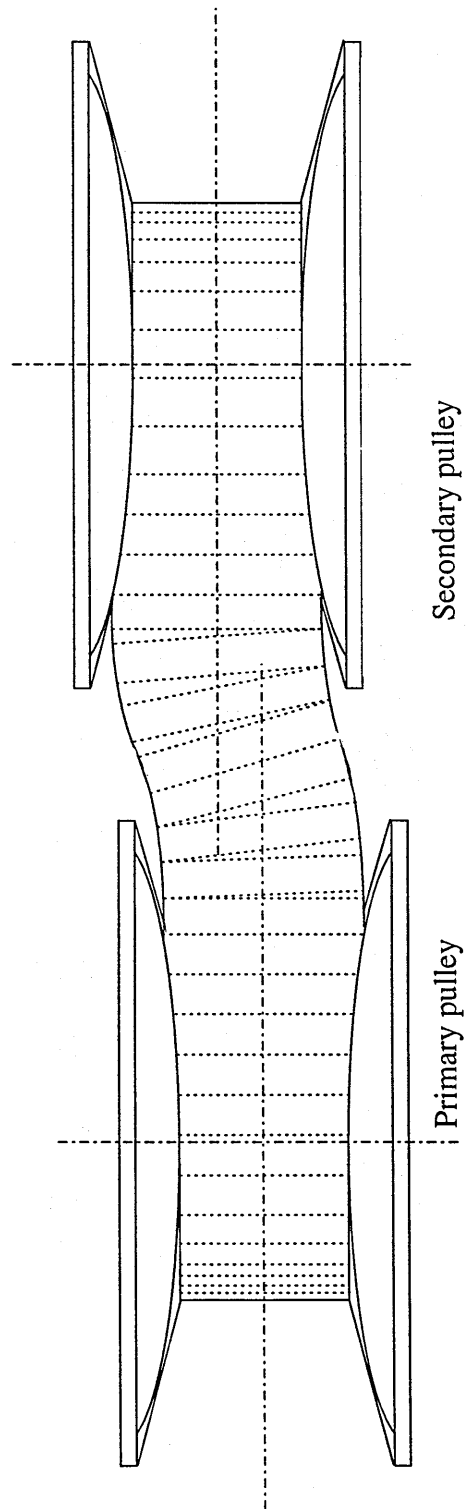
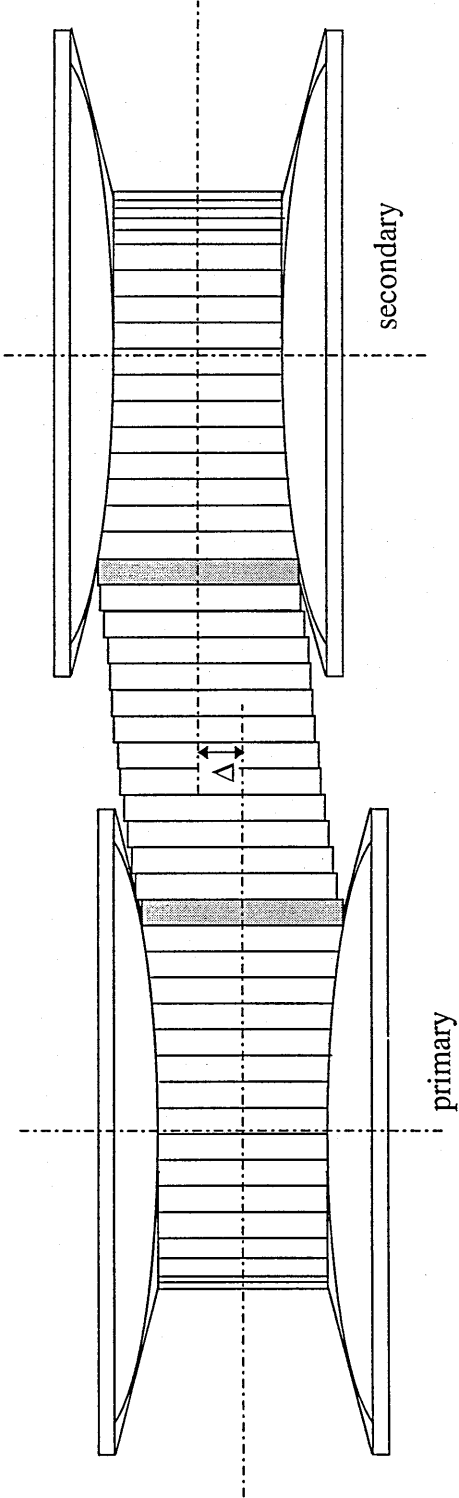


Fig. 4.5 Third and the Proposed Belt Misalignment Model



CHAPTER 5

DERIVATION OF THEORETICAL EQUATION TO CALCULATE THE VAN DOORNE METAL PUSHING V-BELT CVT EFFICIENCY DUE TO BELT MISALIGNMENT

5.1 Introduction

For convenience, the metal pushing V-belt CVT is divided into 4 sections and the bands are considered to be passing over a continuous set of segments around the primary and secondary pulleys in a clockwise direction as shown in Fig. 5.1. It is assumed that the bands leave the primary pulley at exit point 1 at radius RB_p and then enter the secondary pulley at entry point 2 at radius RB_s . The bands radius is assumed to stay constant at RB_s until they leave the secondary pulley at exit point 3. The bands then re-enter the primary pulley at entry point 4 at radius RB_p and they are assumed to stay at this constant radius until they leave the pulley again at exit point 1. The active angles the belt makes between entry and exit points at the primary and secondary pulleys are β_p and β_s respectively. Similarly, the compression forces C_1 and C_4 are assumed to be acting at radius RA_p , while C_2 and C_3 act at radius RA_s .

5.2 Theoretical Assumptions

5.2.1 Main Assumptions

The following assumptions were made for this theoretical analysis model.

- i. The pulley cone surfaces are relatively more rigid than the segments, hence for this analysis, only the segments elasticity are considered.
- ii. The segments are compressed uniformly and equally at both ends.
- iii. The belt leaves and enters the pulleys tangentially.
- iv. The pulley axial compression forces remain relatively constant during operation.
- v. The bands and segments travelling along the compression straight remain relatively plain over the length X' (Fig. 5.2). That is, they do not have a radius of curvature. Thus, friction forces between the bands and the segments shoulders along the compression straight are considered negligible.

- vi. Coulomb friction exists between all the sliding components.
- vii. Only steady state conditions are considered.
- viii. There is no belt slip.

5.2.2 Belt Behaviour

The following are assumed to take place to the segments as they travels around the pulleys and along the two straights (Fig. 5.1):-

Section 4-1

The applied friction forces between the segment sides and the primary pulley cone surfaces that are being developed from the input torque M_p , tend to push the segments forward in the direction of rotation thus compressing the segments gradually along their rocking edges at radius RR_p . The segments compression force then increases correspondingly from zero ($C_4 = 0$) at the entry point 4 and reaches its maximum value C_1 , at exit point 1.

Section 1-2

The belt leaves the primary pulley tangentially and the segments compression force is assumed to remain constant along this straight. That is, $C_2 = C_1$. Along this straight, the compression force acts on the segments contact surfaces at the centre of their contact area. That is, C_1 acts at radius RA_p and C_2 at radius RA_s as shown in Fig. 5.3.

Section 2-3

The opposite then takes place along this section as compared to section 4-1. That is, the belt enters the secondary pulley tangentially. The segments compression force which is now acting along their rocking edges at radius RR_s , reduces correspondingly from $C_2 = C_1$, at entry point 2 to $C_3 = 0$ at exit point 3; at the same time transmitting the output torque M_s to the secondary pulley via the friction forces that are developed between the segment sides and the secondary pulley cone surfaces.

Section 3-4

The segments are ferried back to the primary pulley by the two sets of tension bands and the segments compression force is assumed to remain zero up to the primary pulley entry point 4.

For the bands (the two sets of bands combined together for convenience), the following are assumed to take place as it travels around the pulleys and along the two straights (Fig. 5.1):-

The bands all along in general, travel faster than the segments because they are always at a greater effective radii than the segments [Micklem J. D. 1990], their tension must always be greater than the segments compression force to prevent buckling [Cuypers M. H. 1980] and it is assumed to remain constant all along the two straights. That is, the bands tension at exit point 1 of the primary pulley T_1 , is be equal to the bands tension at entry point 2 of the secondary pulley T_2 , and also has to be greater than the segments compression forces C_1 and C_2 (bearing in mind that $C_2 = C_1$). Similarly along the return straight, where $T_3 = T_4$ and must be greater than zero (since $C_3 = C_4 = 0$). Around both pulleys, bands and segments are assumed to move at the same angular velocity. The bands which are at a larger radius have greater circumferential velocity.

5.3 Bands Tension Equation

Consider a small section of the belt as shown in Fig. 5.3 and 5.4, the element aa' subtends an angle $\delta\vartheta$ around the primary pulley. In general, as mentioned earlier the bands travel faster than the segments at steady state condition due to their greater effective radius. This in turn, results in the friction force between the bands and the segments shoulders $\mu \cdot \delta N_{BP}$, opposing the bands motion which is relatively faster than the segments. Thus, the bands tension is T at a and $T + \delta T$ at a' . Besides these forces, the forces acting on the element are the reaction of the segments δN_{BP} and the bands centrifugal force δU_{BP} . Assuming that the belt acts as a solid body when travelling around the pulleys [Micklem J. D. et al 1994b], there will not be any

slipping between the bands and segments and assuming that there is no slipping between the segments and the pulleys, the bands centrifugal force δU_{BP} , is given by,

$$\delta U_{BP} = m_B \cdot \omega_P^2 \cdot RB_P^2 \cdot \delta \vartheta$$

Resolving forces in the horizontal plane,

$$\sum F_x = 0 :$$

$$\left(\delta T \cdot \cos \frac{\delta \vartheta}{2} \right) - \mu \cdot \delta N_{BP} = 0 \quad (5.3.1)$$

Resolving forces in the vertical plane,

$$\sum F_y = 0 :$$

$$\delta U_{BP} + \delta N_{BP} - (2T + \delta T) \cdot \sin \frac{\delta \vartheta}{2} = 0 \quad (5.3.2)$$

Solving equation (5.3.1) for δN_{BP} and substituting it into equation (5.3.2) we obtain after reduction,

$$\delta U_{BP} - (2T + \delta T) \sin \frac{\delta \vartheta}{2} + \frac{\delta T}{\mu} \cdot \cos \frac{\delta \vartheta}{2} = 0$$

Substituting δU_{BP} in terms of $\delta \vartheta$ and divide the whole equation by $\delta \vartheta$, we obtain,

$$m_B \cdot \omega_P^2 \cdot RB_P^2 \cdot \frac{\delta \vartheta}{\delta \vartheta} - \left\{ \left(T + \frac{\delta T}{2} \right) \cdot \left(\frac{\sin \frac{\delta \vartheta}{2}}{\frac{\delta \vartheta}{2}} \right) \right\} + \frac{1}{\mu} \cdot \frac{\delta T}{\delta \vartheta} \cdot \cos \frac{\delta \vartheta}{2} = 0$$

As $\delta \vartheta$ tends to zero and ignoring small terms, we obtain,

$$\frac{dT}{T - m_B \cdot \omega_P^2 \cdot RB_P^2} = \mu \cdot d\vartheta$$

Integrating the equation from a_4 to a_1 , at a_4 , $T = T_4$ and $\vartheta = 0$; at a_1 , $T = T_1$ and $\vartheta = \beta_P$, we obtain,

$$\int_{T_4}^{T_1} \frac{dT}{T - m_B \cdot \omega_P^2 \cdot RB_P^2} = \mu \int_0^{\beta_P} d\vartheta$$

After simplifying we get,

$$\frac{T_1 - m_B \cdot \omega_P^2 \cdot RB_P^2}{T_4 - m_B \cdot \omega_P^2 \cdot RB_P^2} = e^{\mu \beta_P} \quad (5.3.3)$$

5.4 Segments Compression Equation

For the purpose of analysis, the segments were considered to be continuous around the pulleys [Micklem J. D. 1990]. Consider a small section of the segments as shown in Fig. 5.3, 5.5 and 5.6, element aa' subtending an angle $\delta\vartheta$ around the primary pulley. The segments compression force is C at a and $C + \delta C$ at a' due to the fact that the torque is pushing forward in the direction of rotation of the segments (thus, the increase of the segments compression force by δC at a'). For the torque to be transferred effectively, there must not be any tangential slipping between the segment sides and the pulley cone surfaces. For this to happen, the pulley axial compression force must be designed in such a way that the corresponding friction force developed between the mentioned interface should be great enough to prevent tangential slipping from taking place when the engine or input torque M_p , is being applied. That is, the friction force δW_p acting at radius RC_p (Fig. 5.5), resulted from the input torque M_p , must be less than or equal to the friction force available (in the tangential direction) between the two surfaces. Assuming that the torque is uniformly distributed around the pulley then,

$$\delta W_p = \frac{M_p}{RC_p} \cdot \frac{\delta\vartheta}{\beta_p} \quad (5.4.1)$$

The Van Doorne metal pushing V-belt CVT however, must also be designed in such a way that the belt should be able to slide radially in and out of the pulley cones relatively smoothly with the minimum friction force possible. Hence, the small cone angle of ($\theta = 11^\circ$) and the belt being fully lubricated so that the friction angle ϕ , is less than the pulley-cone angle θ . That is, the segments can slide back in radially to their lowest position at the primary pulley under the bands tension alone. The bands also govern the radial movement of the segments and act as a speed ratio controller. Due to these effects, the reaction force δN_p , on the segment sides resulting from the applied pulley axial compression force δF_p , are not normal but act at the friction angle ϕ , to the segment sides as shown in Figures 5.5 and 5.6. Besides these forces, the forces acting on the element are the equal and opposite friction force $\mu \cdot \delta N_{BP}$, between the bands and the segments shoulders, and its centrifugal force δU_{segP} . Where,

$$\delta U_{segP} = m_s \cdot \omega_p^2 \cdot RR_p^2 \cdot \delta \vartheta \quad (5.4.2)$$

Assuming that the adjustable primary pulley slides smoothly (no friction) on its roller bearing (Fig. 5.6), the reaction force of the pulley δQ_p , acts normal to the pulley shaft. Thus, the reaction force of the segments towards the pulley cone surfaces δN_p , is given by,

$$\delta N_p = \frac{\delta F_p}{\cos(\theta - \phi)} = \frac{F_p \cdot \delta \vartheta}{\beta_p \cdot \cos(\theta - \phi)} \quad (5.4.3)$$

where,

$$\delta F_p = F_p \cdot \frac{\delta \vartheta}{\beta_p} \quad (5.4.4)$$

Denoting the normal component of force on the primary pulley cone surfaces by $\delta N'_p$, we obtain,

$$\delta N'_p = \delta N_p \cdot \cos \phi \quad (5.4.5)$$

Thus, δW_p must be less than or equal to $\mu \cdot \delta N'_p$, to prevent tangential slipping between the primary pulley cone surfaces and the segment sides.

Resolving forces in the horizontal plane (Fig. 5.5),

$$\sum F_x = 0 :$$

$$-\delta C \cdot \cos \frac{\delta \vartheta}{2} + \mu \cdot \delta N_{BP} + \delta W_p = 0 \quad (5.4.6)$$

Resolving forces in the vertical plane,

$$\sum F_y = 0 :$$

$$(2C + \delta C) \cdot \sin \frac{\delta \vartheta}{2} + 2 \cdot \delta N_p \cdot \sin(\theta - \phi) + \delta U_{segP} - \delta N_{BP} = 0 \quad (5.4.7)$$

Proceeding as before (as in section 5.3), we obtain,

$$\frac{C_1 + A}{C_4 + A} = e^{\mu \beta_p} \quad (5.4.8)$$

where,

$$A = \frac{2 \cdot F_p \cdot \tan(\theta - \phi)}{\beta_p} + \frac{M_p}{\mu \cdot RC_p \cdot \beta_p} + m_s \cdot \omega_p^2 \cdot RR_p^2 \quad (5.4.9)$$

since $C_4 = 0$, we obtain,

$$C_1 = A \cdot (e^{\mu\beta_p} - 1) \quad (5.4.10)$$

This is the value of the segments compression force along the compression straight. (Note:- A is always positive and the terms in the brackets also will always be positive thus, C_1 will always be positive).

5.5 Van Doorne Metal Pushing V-belt CVT Geometrical Expressions

Fig. 5.7 shows a detail schematic diagram of the author's proposed belt misalignment model, with belt misalignment Δ . RCF_p is the radius of curvature of the cross-sectional-area of the primary pulley cone surface at the centre of the contact surface between the segments and the pulley cone surfaces (Fig. 5.8). Strictly speaking, this curve is a parabola, but since the length of arc concerned at the pulley exit is relatively small for this particular analysis, it should be sufficiently accurate to assume that RCF_p or RCF_s (at the secondary pulley) is constant and equal to the radius of curvature of the actual surface. By using Unigraphics Ver. 10.4.0 Solid Modeller, the author has produced a table of values of the belt radii R , of the pulley against the corresponding radii of curvatures for pulley cone surface with a wedge angle of 11° , RF , as shown in Table 5.1 and Fig. 5.9. It shows that the relationship between them is linear and can be represented by ,

$$RF = K_f \cdot R - K_{f1} \quad (5.5.1)$$

with the constants $K_f = 5.3389$ (gradient of the slope) and $K_{f1} = 2.2031$ (intersection on the RF axis. This then is the general equation for the relationship between the belt radii and the corresponding pulley cone surface radii of curvature.

However, the relationship between RB_p and RB_s is not linear (Fig. 5.9). The values of RB_s relative to RB_p (Fig. 3.1) can be deduced from equations (3.2.5) and (3.2.6), that is,

$$RB_p = \frac{[L - X \cdot \beta_s \cdot \sin \alpha - 2 \cdot X \cdot \sqrt{(1 - \sin^2 \alpha)}]}{2 \cdot \pi} \quad (5.5.2)$$

and,

$$RB_s = X \cdot \sin \alpha + RB_p \quad (5.5.3)$$

where,

$$\beta_p = \pi - 2 \cdot \alpha \quad (5.5.4)$$

and,

$$\beta_s = \pi + 2 \cdot \alpha \quad (5.5.5)$$

Thus, from Fig. 5.8 and 5.10 the respective primary pulley belt radii is given by,

$$RC_p = RB_p - \frac{t_B}{2} - q - \bar{y} \cdot \cos\theta \quad (5.5.6)$$

$$RR_p = RB_p - \frac{t_B}{2} - q \quad (5.5.7)$$

and,

$$RD_p = RB_p - \frac{t_B}{2} \quad (5.5.8)$$

where,

$$\bar{y} = \frac{c \cdot (a + 2 \cdot b)}{3 \cdot (a + b)} \quad (5.5.9)$$

Correspondingly, the primary pulley cone radii of curvature is given by,

$$RCF_p = K_f \cdot RC_p - K_{f1} \quad (5.5.10)$$

$$RRF_p = K_f \cdot RR_p - K_{f1} \quad (5.5.11)$$

$$RDF_p = K_f \cdot RD_p - K_{f1} \quad (5.5.12)$$

and,

$$RBF_p = K_f \cdot RB_p - K_{f1} \quad (5.5.13)$$

Similar procedures apply to the secondary pulley.

By referring back to Fig. 5.7, let points $O_p(x_{OP}, y_{OP})$ and $O'_p(x'_{OP}, y'_{OP})$ be the centre of curvature of the cross sectional area of the top and bottom halves of the primary pulley cones, respectively and O''_p be the mid-point between them. Note also that, point $O_p(x_{OP}, y_{OP})$ is the origin of the axes with positive x_p to the right and positive y_p vertically downwards. Let lines a_p and b_p represent the edges of the segments when they are free from any side compression forces, and curves a_p and b_p represent the pulley cone surface; while points $a_p(x_{ap}, y_{ap})$ and $b_p(x_{bp}, y_{bp})$ be the intersection

points between curve a_p and line a_p and, curve b_p and line b_p , respectively. Now, let $c_p(x_p, y_p)$ be any point on the curve a_p , thus from trigonometry, line a_p can be represented by y_{ap} and curve a_p by equation,

$$x_p^2 + y_p^2 = RCF_p^2 \quad (5.5.14)$$

where,

$$y_{ap} = RCF_p - \frac{\delta d_p}{2} \quad (5.5.15)$$

From standard engineering stress-strain equations, the segments strain at the primary pulley ϵ_{segP} , is given by,

$$\epsilon_{segP} = \frac{\sigma_{segP}}{E_{seg}} = \frac{\delta d_p}{d} = \frac{d - d'_p}{d} = \frac{\frac{F_{segP}}{A_{seg}}}{E_{seg}} \quad (5.5.16)$$

and by taking β_p to be the angle of lap and neglecting end effects, the number of segments clamped between the primary pulley is equal to $\left(\frac{RR_p \cdot \beta_p}{a}\right)$ and the axial compression force per segment F_{segP} , is given by,

$$F_{segP} = \frac{F_p}{\left(\frac{RR_p \cdot \beta_p}{a}\right)} \quad (5.5.17)$$

and the effective cross-sectional area of the segment A_{seg} as shown in Fig. 5.10, is given by,

$$A_{seg} = \frac{1}{2} \cdot (a + b) \cdot c \cdot \cos \theta \quad (5.5.18)$$

Thus, from equation (5.5.16) we can calculate d'_p and δd_p . Where,

$$\delta d_p = \frac{F_{segP} \cdot d}{E_{seg} \cdot A_{seg}} \quad (5.5.19)$$

and,

$$d'_p = d - \delta d_p \quad (5.5.20)$$

where,

$$d = d_m - 2 \cdot (\bar{y} \cdot \sin \theta) \quad (5.5.21)$$

Consequently, y_{ap} can then be calculated from equation (5.5.15).

Knowing y_{aP} , x_{aP} can then be calculated by substituting y_{aP} into equation (5.5.14) where,

$$x_{aP} = +\sqrt{(RCF_P^2 - y_{aP}^2)} \quad (5.5.22)$$

and $x_{bP} = x_{aP}$. For the secondary pulley, the same procedure applies, except that x_{aS} and x_{bS} have negative values.

From the magnitude of x_{aP} and x_{aS} values at the primary and secondary pulleys respectively, the clamping distance of the segments along the compression straight X' , can now be calculated (Fig. 5.2). That is,

$$X' = X \cdot \cos \alpha - \{|x_{aP}| + |x_{aS}|\} \quad (5.5.23)$$

Thus, the number of segments clamped between the 2 pulleys, neglecting the segments lateral deformation n , is equal to X'/a . Hence from here, we should be able to calculate the work done, then the rate of work done and hence power loss by the transmission due to belt misalignment.

5.6 Transmission Efficiency

From Fig. 5.1, taking moments about the axis of rotation of the primary pulley at point O_P ,

$$\sum M_{O_P} = 0: \quad (T_4 - T_1) \cdot RB_P + (C_1 - C_4) \cdot RA_P - M_P = 0 \quad (5.6.1)$$

But $C_4 = 0$, thus,

$$(T_4 - T_1) \cdot RB_P + C_1 \cdot RA_P - M_P = 0 \quad (5.6.2)$$

Similarly at the secondary pulley (note:- $T_1 = T_2$ and $T_3 = T_4$),

$$(T_4 - T_1) \cdot RB_S + C_1 \cdot RA_S - M_S = 0 \quad (5.6.3)$$

Manipulating equations (5.3.3) and (5.6.2), we obtain,

$$T_4 = (m_B \cdot \omega_P^2 \cdot RB_P^2) + \frac{\left(\frac{C_1 \cdot RA_P - M_P}{RB_P} \right)}{(e^{\mu\beta_P} - 1)} \quad (5.6.4)$$

and,

$$T_1 = e^{\mu\beta_r} \left\{ T_4 - (m_B \cdot \omega_P^2 \cdot RB_P^2) \right\} + m_B \cdot \omega_P^2 \cdot RB_P^2 \quad (5.6.5)$$

Divide through equations (5.6.2) and (5.6.3) by RB_P and RB_S respectively and then subtract one from the other, we obtain after rearrangement,

$$M_S = M_P \cdot \frac{RB_S}{RB_P} + RB_S \cdot C_1 \left(\frac{RA_S}{RB_S} - \frac{RA_P}{RB_P} \right) \quad (5.6.6)$$

This equation represents the ideal secondary output pulley torque. Also, from this equation we can see that if the bands tension and the segments compression forces coincide (that is, $RB_P = RA_P$ and $RB_S = RA_S$), the term in the brackets is reduced to zero. Thus, the ideal output torque M_S , is then just a function of the bands effective radii only.

Ideally, the input power H_P , into the primary pulley is equal to the output power H_S , given out by the secondary pulley. That is,

$$H_P = H_S \quad (5.6.7)$$

where,

$$H_P = M_P \cdot \omega_P \quad \text{and} \quad H_S = M_S \cdot \omega_S \quad (5.6.8)$$

Thus,

$$\omega_S = \frac{M_P \cdot \omega_P}{M_S} \quad (5.6.9)$$

However practically, the input power H_P , into the primary pulley is equal to the actual output power H'_S , given out by the secondary pulley, plus the total power loss by the belt TPL . That is,

$$H_P = H'_S + TPL \quad (5.6.10)$$

or,

$$M_P \cdot \omega_P = M'_S \cdot \omega_S + TPL \quad (5.6.11)$$

where, M'_S is the actual output torque at the secondary pulley. Thus,

$$M'_S = \frac{M_P \cdot \omega_P}{\omega_S} - \frac{TPL}{\omega_S} \quad (5.6.12)$$

or,

$$M'_S = M_S - \frac{TPL}{\omega_S} \quad (5.6.13)$$

That is, practically the actual output torque M'_S , is equal to the ideal output torque M_S , less the total power loss by the belt TPL divided by the output speed ω_S . Assuming that the bands tension and segments compression all along the compression straight coincide [Micklem J. D. et al 1994a, Fig. 3], equation (5.6.4) becomes,

$$T_4 = (m_B \cdot \omega_P^2 \cdot RB_P^2) + \frac{\left(\frac{C_1 \cdot RB_P - M_P}{RB_P} \right)}{(e^{\mu\beta_P} - 1)} \quad (5.6.14)$$

and equation (5.6.6) becomes,

$$M_S = M_P \cdot \frac{RB_S}{RB_P} \quad (5.6.15)$$

while equation (5.6.5) remains the same. Thus, equation (5.6.9) becomes,

$$\omega_S = \frac{\omega_P}{\left(\frac{RB_S}{RB_P} \right)} = \frac{\omega_P}{i} \quad (5.6.16)$$

where,

$$i = \frac{RB_S}{RB_P} \quad (5.6.17)$$

that is, the belt speed ratio; and equation (5.6.13) becomes,

$$M'_S = M_S \left(1 - \frac{TPL}{H_P} \right) \quad (5.6.18)$$

This is the actual secondary output pulley torque after considering the total power loss of the belt.

This shows that, the actual output torque M'_S , is equal to zero if the total power loss by the belt TPL , is equivalent to the input power (that is, the term in the brackets reduces to zero); and if the total power loss by the belt TPL , is very small or negligible, the actual output torque M'_S , is the same as the ideal output torque M_S , (that is, the term in the brackets reduces to unity). Thus, by substituting the appropriate total power loss value into the above equation, the transmission belt efficiency η , can be calculated as follows,

$$\eta = \frac{\text{actual output torque}}{\text{ideal output torque}} * 100\%$$

That is,

$$\eta = \frac{M'_s}{M_s} * 100\% \quad (5.6.19)$$

The percentage of output torque loss $ML_s\%$, is given by,

$$ML_s\% = \left(1 - \frac{\text{Actual output torque}}{\text{Ideal output torque}} \right) * 100\%$$

That is,

$$ML_s\% = \left(1 - \frac{M'_s}{M_s} \right) * 100\% \quad (5.6.20)$$

5.7 Power Loss Due To Compressed Segments Sliding Against Pulley Cone Surfaces During Exit From And Entry Into The Pulley Grooves

Due to the pulleys clamping action, during entry into and exit from the primary and secondary pulleys some work is done by the metal pushing V-belt to overcome the sliding friction between the segment sides and the pulley cone surfaces. This results in some power loss in the transmission belt. During this phase, the axial load exerted on the pulley cones gives compression loading on the segments. The segments deflect (are compressed) by the compression loading F_p as shown in Fig. 5.11. Fig. 5.12 shows the path of the segments sliding against the primary pulley cone surface during exit, point $a_p(x_{ap}, z_{ap})$ is the take off point where the segments recover their full free length d . For analysis purposes, sliding is assumed to take place at the centre of the sliding contact area between the segments and the pulley cone surfaces at radius RC_p (Fig. 5.12). Curve a , as shown in Fig. 5.12, represents the locus of the centre of contact area on the primary pulley cone surface with radius RC_p . Assuming that the belt leaves the pulleys tangentially and no tangential slipping takes place between the segments and the pulleys, the sliding between the segment sides and the pulley cone surfaces describes an involute profile $a_p b_p$ or $c_p d_p$ etc. However, since the value of x_{ap} is relatively small, it is reasonably accurate to assume that this involute profile to

be equivalent to an arc with radius of curvature x_{aP} , which subtends an angle $\frac{\phi_{aP}}{2}$ and with an arc length of l_{bP} , where l_{bP} is the sliding locus and in this case is also the total sliding distance between the segment sides and the pulley cone surfaces.

From Fig. 5.12, angle ϕ_{aP} is equal to

$$\phi_{aP} = 2 \cdot \sin^{-1} \frac{x_{aP}}{2 \cdot RCF_p} \quad (5.7.1)$$

and,

$$l_{bP} = x_{aP} \cdot \frac{\phi_{aP}}{2} \quad (5.7.2)$$

Fig. 5.13 shows the plan view of Fig. 5.12 with curve b representing the primary pulley cone surface outline at radius RCF_p . Assuming as before that since the arc concerned is very small, it is reasonably accurate to assume that curve b is a circle with radius RCF_p . Point $c_p (x, y)$ represents a general point on curve b . Assuming that the segments deflect uniformly and equally at both ends,

$$\frac{\delta d'_p}{2} = \text{functions of } RR_p, RCF_p \text{ and angle } \phi_{aP}$$

Curve b can be represented by equation,

$$y = \sqrt{RCF_p^2 - x^2} \quad (5.7.3)$$

and, y_{aP} is given by equation (5.5.15) while,

$$\delta d'_p = 2(y - y_{aP}) \quad (5.7.4)$$

where,

$$y_{aP} \leq y \leq RCF_p \quad (5.7.5)$$

Substituting y into $\delta d'_p$, we get,

$$\delta d'_p = 2\left(\sqrt{RCF_p^2 - x^2} - y_{aP}\right) \quad (5.7.6)$$

Neglecting end effects and assuming that the clamping force per segment in this case is directly proportional to the ratio of the segments deflection over its total deflection along the x-axis, the axial compression force per segment F'_{segP} from the tangency point O' to the take off point a_p , is given by,

$$F'_{segP} = K_P \cdot \frac{\delta d'_P}{\delta d_P} = K_P \cdot \frac{2(\sqrt{RCF_P^2 - x^2} - y_{aP})}{\delta d_P} \quad (5.7.7)$$

where, K_P is a constant with units of $\frac{N}{seg}$. In this case, the magnitude of K_P has the value of F'_{segP} as given by equation (5.5.17). Thus,

$$F'_{segP} = \frac{F_P}{\left(\frac{RR_P \cdot \beta_P}{a}\right)} \cdot \frac{2(\sqrt{RRF_P^2 - x^2} - y_{aP})}{\delta d_P} \quad (5.7.8)$$

F'_{segP} is then the force per segment that is responsible for the frictional losses between the segment sides and the pulley cone surfaces during exit and entry into the primary pulley.

From Fig. 5.12, it can be seen that the sliding, which takes place between the segment sides and the pulley cone surfaces begins, at the tangency point O' and continues to the take off point a_P , a distance of l_{bP} . Now, consider an elemental length δl , representing the small sliding locus at distance δx from the tangency point O' (Fig. 5.12 and 5.13) where,

$$\delta l = \frac{\delta x}{x_{aP}} \cdot l_{bP} = \frac{\delta x}{x_{aP}} \cdot \left(x_{aP} \cdot \frac{\phi_{aP}}{2}\right) = \frac{\phi_{aP}}{2} \cdot \delta x \quad (5.7.9)$$

Thus, the elemental work done by one segment (note the 2 ends per segment) δU_{1P} , as the segments travel from the tangency point O' to the take off point a_P , along the x-axis is given by,

$$\delta U_{1P} = 2 \cdot \mu \cdot F'_{segP} \cdot \delta l \quad (5.7.10)$$

Substituting F'_{segP} and δl we get,

$$\delta U_{1P} = 2 \cdot \mu \cdot \left[\frac{F_P}{\left(\frac{RR_P \cdot \beta_P}{a}\right)} \cdot \frac{2(\sqrt{RCF_P^2 - x^2} - y_{aP})}{\delta d_P} \right] \cdot \left(\frac{\phi_{aP}}{2} \cdot \delta x\right) \quad (5.7.11)$$

Rearrange we get,

$$\delta U_{1P} = \frac{2 \cdot \mu \cdot F_P \cdot a \cdot \phi_{aP}}{RR_P \cdot \beta_P \cdot \delta d_P} \cdot (\sqrt{RCF_P^2 - x^2} - y_{aP}) \cdot \delta x \quad (5.7.12)$$

Integrating $\delta U1_p$ from $x=0$ to $x=x_{aP}$, we will then get the total work done by one segment during exit from the primary pulley $U1_p$. That is,

$$U1_p = \frac{2 \cdot \mu \cdot F_p \cdot a \cdot \phi_{aP}}{RR_p \cdot \beta_p \cdot \delta d_p} \cdot \int_0^{x_{aP}} \left(\sqrt{RCF_p^2 - x^2} - y_{aP} \right) dx_p \quad (5.7.13)$$

Thus, $U1_p = \frac{2 \cdot \mu \cdot F_p \cdot a \cdot \phi_{aP}}{RR_p \cdot \beta_p \cdot \delta d_p} *$

$$\left[\frac{RCF_p^2}{2} \left\{ \left(\sin^{-1} \frac{x_{aP}}{RCF_p} \right) + \left(\frac{x_{aP} \sqrt{RCF_p^2 - x_{aP}^2}}{RCF_p^2} \right) \right\} - x_{aP} \cdot y_{aP} \right] \quad (5.7.14)$$

Therefore, the power loss due to segments sliding against the pulley cone surfaces during exit from the primary pulley, is given by the total work done per segment $U1_p$, multiplied by the number of segments passing per second v_{segP} . Where,

$$v_{segP} = \omega_p \cdot \frac{RR_p}{a} \left[\frac{seg}{s} \right] \quad (5.7.15)$$

Thus, the power loss due to segments sliding between the segment sides and the pulley cone surfaces during exit from the primary pulley $(PL_{sl})_{Exp}$ is given by,

$$(PL_{sl})_{Exp} = U1_p \cdot v_{segP} \quad (5.7.16)$$

That is,

$$(PL_{sl})_{Exp} = \frac{2 \cdot \mu \cdot F_p \cdot \omega_p \cdot \phi_{aP}}{\beta_p \cdot \delta d_p} * \left[\frac{RCF_p^2}{2} \left\{ \left(\sin^{-1} \frac{x_{aP}}{RCF_p} \right) + \left(\frac{x_{aP} \sqrt{RCF_p^2 - x_{aP}^2}}{RCF_p^2} \right) \right\} - x_{aP} \cdot y_{aP} \right] \quad (5.7.17)$$

Assuming that similar happenings take place when the segments are entering the primary pulley, the total power loss due to segments sliding at the primary pulley during entry and exit PL_{slP} , is equal to twice the loss at exit from the primary pulley.

That is,

$$PL_{slP} = 2(PL_{sl})_{Exp} \quad (5.7.18)$$

That is,

$$PL_{slP} = \frac{4 \cdot \mu \cdot F_P \cdot \omega_P \cdot \phi_{aP}}{\beta_P \cdot \delta d_P} * \left[\frac{RCF_P^2}{2} \left\{ \left(\sin^{-1} \frac{x_{aP}}{RCF_P} \right) + \left(\frac{x_{aP} \sqrt{RCF_P^2 - x_{aP}^2}}{RCF_P^2} \right) \right\} - x_{aP} \cdot y_{aP} \right] \quad (5.7.19)$$

Assuming that similar happenings also take place at the secondary pulley, thus the power loss due to segments sliding during entry and exit into and out of the secondary pulley PL_{slS} , is given by,

$$PL_{slS} = \frac{4 \cdot \mu \cdot F_S \cdot \omega_S \cdot \phi_{aS}}{\beta_S \cdot \delta d_S} * \left[\frac{RCF_S^2}{2} \left\{ \left(\sin^{-1} \frac{x_{aS}}{RCF_S} \right) + \left(\frac{x_{aS} \sqrt{RCF_S^2 - x_{aS}^2}}{RCF_S^2} \right) \right\} - x_{aS} \cdot y_{aS} \right] \quad (5.7.20)$$

Hence, the total power loss due to segments sliding during exit and entry PL_{sl} , is equal to,

$$PL_{sl} = PL_{slP} + PL_{slS} \quad (5.7.21)$$

This is the power loss in the belt system due to entry and exit of the segments to the two pulleys.

Note:- The power loss due to this phenomena is independent of belt misalignment. That is, the losses are always present in the transmission during operation.

5.8 Power Loss Due To Belt Misalignment

5.8.1 Introduction

The Van Doorne metal pushing V-belt CVT has been designed in such a way that the bands width is smaller than the segment shoulder width. This, allows the bands to slide sideways on the segment shoulders. Due to this effect, at steady state, it is assumed that the two sets of bands realign themselves without touching the segment central columns and the pulley cone surfaces. However, if the magnitude of the belt misalignment Δ , exceeds the allowable belt misalignment limits e_p or e_s , on the primary or secondary pulleys respectively as shown in Fig. 5.14, the bands are pushed against the segment central columns by the pulley cone surfaces.

For example, if the belt misalignment is as shown as in Fig. 5.14, both the adjustable halves of the primary and secondary pulley cone surfaces will push the two sets of bands transversely and accordingly against the segment shoulders. The first set of bands rests against the central columns of the segments near the fixed halves of the primary pulley, while the other set is pushed against the segment central columns near the fixed halves of the secondary pulley. Due to the curvature of the pulley cone surfaces, the bands bend accordingly. The reverse will take place when the belt misalignment is reversed. This then results in sliding between the bands sides and the pulley cone surfaces which in turn leads to power loss. At the primary pulley, the limit of this excessive belt misalignment takes place when the belt misalignment Δ , begins to exceed the minimum allowable belt misalignment limit e_p (Fig 5.8), where,

$$e_p = \left\{ \frac{d'_p}{2} - \left(d_B + \frac{f}{2} \right) \right\} + q \tan \theta \quad (5.8.1)$$

Similarly, at the secondary pulley the excessive belt misalignment limit e_s , is given by,

$$e_s = \left\{ \frac{d'_s}{2} - \left(d_B + \frac{f}{2} \right) \right\} + q \tan \theta \quad (5.8.2)$$

Thus, when considering power loss due to belt misalignment, one must take into consideration whether the allowable limits are being exceeded or not. During small belt misalignment, the only sliding that is taking place is the sliding between the segment front and back faces along the compression straight. Sliding should also take

place along the return side, but since the segment compression force along the return straight is zero, no power loss will take place along the return straight and it is being neglected in this analysis. In short, for small belt misalignment, the belt misalignment Δ , should be less than the allowable belt misalignment limits e_p or e_s and the only sliding that will be taking place will be between the back and the front faces of the segments themselves, as has been explained in the author's proposed belt misalignment model (Chapter 4).

Excessive belt misalignment means that the transmission belt misalignment Δ , is equal to or greater than the allowable belt misalignment limits e_p or e_s on the primary or secondary pulleys respectively. Thus, the total power loss due to excessive belt misalignment has to take into consideration the power loss due to small belt misalignment as well. Fig. 5.14 shows that, during excessive belt misalignment, the two sets of bands lean towards the pulley cone surfaces. The bands are tangent to the pulley cone surfaces and subscribe an angle φ_e from the tangency points. With some geometrical algebra it can be shown that, the angle φ_e is given by,

$$\varphi_e = \tan^{-1} \left\{ \frac{-l_a l_b - \sqrt{l_a^2 l_b^2 - [l_a^2 - (R_a + R_b)^2][l_b^2 - (R_a + R_b)^2]}}{[l_a^2 - (R_a + R_b)^2]} \right\}^{-1} \quad (5.8.3)$$

where,

$$l_a = RBF_p + d'_p + RBF_s - |\Delta| \quad (5.8.4)$$

$$l_b = X \cdot \cos \alpha \quad (5.8.5)$$

$$R_a = RBF_p \quad (5.8.6)$$

$$R_b = RBF_s + \frac{d'_s}{2} + \frac{f}{2} + d_b \quad (5.8.7)$$

Thus, the force per unit length exerted by the bands to the pulley cone surfaces will be governed by this angle φ_e .

5.8.2 Small Belt Misalignment

Fig. 5.7 shows that if there are n numbers of segments being compressed between the two pulleys along the compression straight, then there are $n - 1$ numbers of surfaces

that experience sliding of equal amount. When leaving the primary pulley, the segments slide out and when entering the secondary pulley, the segments slide in back. Along the compression straight (Fig. 5.15), the segment compression force C_1 is the force that is responsible for the sliding frictional losses between the segment front and back faces. Assuming that the segments slide out uniformly, l_Δ will be the sliding distance per segment. Similarly, during entering into the secondary pulley, the segments slide in back into the secondary pulley groove. Hence, the total sliding distance covered by one segment is twice l_Δ , and the magnitude of l_Δ is given by,

$$l_\Delta = \frac{\Delta}{n-1} \quad (5.8.8)$$

where,

$$n = \frac{X'}{a} \quad (5.8.9)$$

Therefore,

$$l_\Delta = \frac{\Delta}{\left(\frac{X'}{a} - 1\right)} \quad (5.8.10)$$

Thus, the work done by one segment U_{seg} , is equal to,

$$U_{seg} = \mu \cdot C_1 \cdot 2 \cdot l_\Delta \quad (5.8.11)$$

Substituting l_Δ we get,

$$U_{seg} = \mu \cdot C_1 \cdot \frac{2 \cdot \Delta}{\left(\frac{X'}{a} - 1\right)} \quad (5.8.12)$$

By multiplying the work done per segment U_{seg} by the number of segments passing along the compression straight per second v_{segP} , we get the power loss due to the segments sliding against each other on their back and front faces as they pass through the compression straight PL_{sml} . That is,

$$PL_{sml} = U_{seg} \cdot v_{segP} \quad (5.8.13)$$

where,

$$v_{segP} = \omega_P \cdot \frac{RB_P}{a} \quad (5.8.14)$$

Substituting U_{seg} and v_{segP} into PL_{sml} we get,

$$PL_{smi} = \mu \cdot C_1 \cdot \frac{2\Delta}{\left(\frac{X'}{a} - 1\right)} \cdot \omega_p \cdot \frac{RB_p}{a} \quad (5.8.15)$$

After simplifying we get,

$$PL_{smi} = \frac{2 \cdot \mu \cdot C_1 \cdot \Delta \cdot \omega_p \cdot RB_p}{(X' - a)} \quad (5.8.16)$$

This is then the equation to calculate the power loss due to small belt misalignment along the compression straight.

Note:- The equation to calculate the power loss due to small belt misalignment along the compression straight can only be applied provided that the belt misalignment Δ , does not exceed the maximum belt misalignment Δ_{max} (Fig. 5.16), allowed by the equation. That is,

$$\Delta_{max} = (n - 1) h \quad (5.8.17)$$

where, h is the tolerance available between the segment pin and hole. If $\Delta > \Delta_{max}$, there will be no further sliding between the segments along the compression straight. This is because, sliding between the segments is hindered by the segment pins and holes as shown in Fig. 5.15. This however, the author leaves it for future work. Furthermore, this is unlikely to occur in this work because the minimum value of Δ_{max} , which is about 3.25 mm (section 4.4) is greater than the maximum value of Δ , which is about 2.70 mm (Fig. 8.3). Hence, it is not relevance in this work.

Also, if the belt misalignment Δ , begins to exceed its minimum allowable limits e_p or e_s at the primary or secondary pulley respectively, as shown in Fig. 5.14, this equation will have to be coupled with the power loss equation due to excessive belt misalignment that will be covered in the next section. Fig. 5.14 also shows that, the segments will not be forced transversely further by the bands along the compression straight. Therefore, the only forces that keep the segments in equilibrium after being forced to slide out from the primary pulley along the compression straight from the resulting couple of $C_1 \cdot l_\Delta$, due to belt misalignment will be the friction force $\mu \cdot C_1$ multiplied by the segment thickness a (Fig. 5.15). This corresponding couple due to

the frictional force $\mu \cdot C_1 \cdot a$, must then be at least equal to or greater than the resulting couple $C_1 \cdot l_\Delta$. That is,

$$\mu \cdot C_1 \cdot a \geq C_1 \cdot l_\Delta \quad (5.8.18)$$

Thus,

$$\mu \geq \frac{l_\Delta}{a} \quad (5.8.19)$$

Now, the minimum number of segments n , along the compression straight are approximately equal to 66. Say, we have a small belt misalignment Δ of 0.5 mm. and $a = 2.00$ mm. Then,

$$l_\Delta = \frac{\Delta}{n-1} = \frac{0.5}{66-1} = \frac{0.5}{65} \approx 0.0077$$

and,

$$\mu \geq \frac{l_\Delta}{a} \geq \frac{0.0077}{2} \geq 0.0038$$

The practical value of μ for lubricated steel on steel is usually about 0.1 [Parker S. P. 1983 and Deik G. 1990], which is about 26 times greater than the required calculated value. Thus, the segments will not slide further due to belt misalignment and they remain in that position along the compression straight unless there is an external force acting on the segment sides. However as shown in Fig. 5.14, with excessive belt misalignment where, the belt misalignment Δ is greater than the minimum allowable belt misalignment limits e_p or e_s , the side forces from the bands tend to push the segments transversely (via the segment central columns) towards one side of the primary pulley cone surface during exit. At the entry into the secondary pulley, the pulley cone tends to push the segments transversely (also via the segment central columns) to the opposite side of the secondary pulley cone surface. Thus, the two sets of bands cause the segments to slide out during exit from the primary pulley and slide in back during entry into the secondary pulley. But once the segments get into the compression straight, they remain in the same position until they begin to enter the secondary pulley where, the two sets of bands now aid the segments to slide in back into the secondary pulley groove. A similar phenomena will take place along the return side.

5.8.3 Excessive Belt Misalignment

5.8.3.1 Bands Sliding Against Pulley Cone Surfaces

Fig. 5.14 shows that, when excessive belt misalignment takes place, only one set of the bands leans against the pulley cone surfaces during exit from or entry into the pulleys. The other set of bands however, leans against the segment central columns which in turn press the segments axially further against the pulley cone surfaces. Both these phenomena, will result in power losses due to sliding friction on the pulley cone surfaces by the bands sides and due to additional axial compression force on the segments which leads to further frictional losses, also due to sliding friction by the segment sides on the pulley cone surfaces. This section deals only with power losses due to the former while the next will deal with the latter.

Assuming that the bands behave like an ordinary flat belt wrapping over a flat ordinary pulley surface, the free body diagram will be as shown in Fig. 5.17. In this case, since the angle φ_e that is being considered is relatively quite small, it should be reasonably accurate to assume that the bands tension remains relatively constant as it has been previously assumed. Consider a small section of the bands of length δl which subtends an angle $\delta\varphi$. The normal force that acts on the elemental bands F_{B12} , is given by,

$$F_{B12} = 2 \cdot \frac{T_1}{2} \cdot \sin\left(\frac{\delta\varphi}{2}\right) = T_1 \cdot \sin\left(\frac{\delta\varphi}{2}\right) \quad (5.8.20)$$

In the limit,

$$\sin\left(\frac{\delta\varphi}{2}\right) = \frac{d\varphi}{2}$$

Thus,

$$F_{B12} = \frac{T_1}{2} \cdot d\varphi \quad (5.8.21)$$

This is the force on the elemental bands that is responsible for the frictional losses between the elemental bands side and the primary pulley cone surface during exit from the primary pulley and during entry into the secondary pulley. Similarly, during entry into the primary pulley and during exit from the secondary pulley, the similar force F_{B34} , is given by,

$$F_{B34} = \frac{T_4}{2} \cdot d\phi \quad (5.8.22)$$

From Fig. 5.17 (b), it can be shown that the sliding bands speed during exit from the primary pulley v_{slP} , is given by,

$$v_{slP} = RBF_P \cdot \phi \cdot \omega_P \quad (5.8.23)$$

Thus, the power on the elemental bands due to excessive sliding during exit from the primary pulley δPL_{ex1} , is given by,

$$\delta PL_{ex1} = \mu \cdot F_{B12} \cdot v_{slP}$$

After the substitution and simplifying we get,

$$\delta PL_{ex1} = \frac{\mu \cdot T_1 \cdot \omega_P \cdot RBF_P \cdot \phi}{2} \cdot d\phi$$

Integrating from $\phi = 0$ to $\phi = \phi_e$, we get the total work done by the bands due to excessive sliding during exit from the primary pulley U_{31} , which is equal to,

$$PL_{ex1} = \frac{\mu \cdot T_1 \cdot \omega_P \cdot RBF_P \cdot \phi_e^2}{4} \quad (5.8.24)$$

This is the power loss due to one set of the bands sliding against the primary pulley cone surface due to excessive belt misalignment during exit from the primary pulley at exit point 1 (Fig. 5.1). Similarly, proceeding as before to all the other three locations (exit point 3 and entry points 2 and 4), we obtained at entry point 2 of the secondary pulley, the power loss due to one set of the bands sliding against the secondary pulley cone surface due to excessive belt misalignment PL_{en2} , is given by,

$$PL_{en2} = \frac{\mu \cdot T_1 \cdot \omega_S \cdot RBF_S \cdot \phi_e^2}{4} \quad (5.8.25)$$

At exit point 3 of the secondary pulley, the power loss due the bands sliding against the secondary pulley cone surface due to excessive belt misalignment PL_{ex3} , is given by,

$$PL_{ex3} = \frac{\mu \cdot T_4 \cdot \omega_s \cdot RBF_s \cdot \phi_e^2}{4} \quad (5.8.26)$$

and at entry point 4, the power loss due to one set of the bands sliding against the primary pulley cone surface due to excessive belt misalignment PL_{en4} , is given by,

$$PL_{en4} = \frac{\mu \cdot T_4 \cdot \omega_p \cdot RBF_p \cdot \phi_e^2}{4} \quad (5.8.27)$$

Thus, the total power loss PL_{bsl} , due the bands sliding against the pulleys cone surfaces due to excessive belt misalignment is given by,

$$PL_{bsl} = PL_{ex1} + PL_{en2} + PL_{ex3} + PL_{en4} \quad (5.8.28)$$

5.8.3.2 Segments Sliding Against Pulley Cone Surfaces

As has been mentioned in the final part of section 5.8.2 and at the beginning of the previous section, excessive belt misalignment results in further sliding of the segments against the pulley cone surfaces (on top of the definite sliding as explained in section 5.7). This is due to the fact that, the second set of bands tends to press further on the segments via the segment central columns against the pulley cone surfaces during exit from and entry into the pulleys, as shown in Fig. 5.14. Due to this phenomena, similar kind of sliding as that of the previous section will take place and hence, similar kind of sliding losses as well.

In the previous case, the losses are due to the sliding between one set of the bands and the pulley cone surfaces. But for this case, the losses will be the sliding between one side of the segment and the pulley cone surfaces. For both cases, the normal force responsible for the sliding losses are the same, that is F_{B12} and F_{B34} as given by equations (5.8.21) and (5.8.22) respectively, except that this time they are acting on the segment central columns. Neglecting the small difference in the radius at where the sliding take place for both cases, it should be reasonably accurate to assume that the losses for both cases are the same.

Thus, the total power loss due to excessive belt misalignment PL_{exc} , is given by,

$$PL_{exc} = 2 \cdot PL_{bsl} \quad (5.8.29)$$

5.9 Belt Total Power Loss (TPL) Due To Segments Sliding During Exit From And Entry Into The Pulley Cone Surfaces And Due To Belt Misalignment

Therefore, the total power loss TPL , as mentioned in the title of this section is given by,

$$TPL = PL_{sl} + PL_{sml} + PL_{exc} \quad (5.9.1)$$

That is, the total power loss of the metal pushing V-belt that is being considered in this work to be equal to, the power loss due to the axially compressed segments around both the primary and secondary pulleys sliding against the respective pulley cone surfaces plus, the power loss due to the segments sliding against each other on their front and back surfaces due to belt misalignment along the compression straight, plus the power loss due to extra sliding of the axially compressed segments at the entry and exit of both the primary and secondary pulleys due to sliding against the respective pulley cone surfaces.

By substituting this value into equation (5.6.18), the belt efficiency from equation (5.6.19) can now be determined.

5.10 Figures

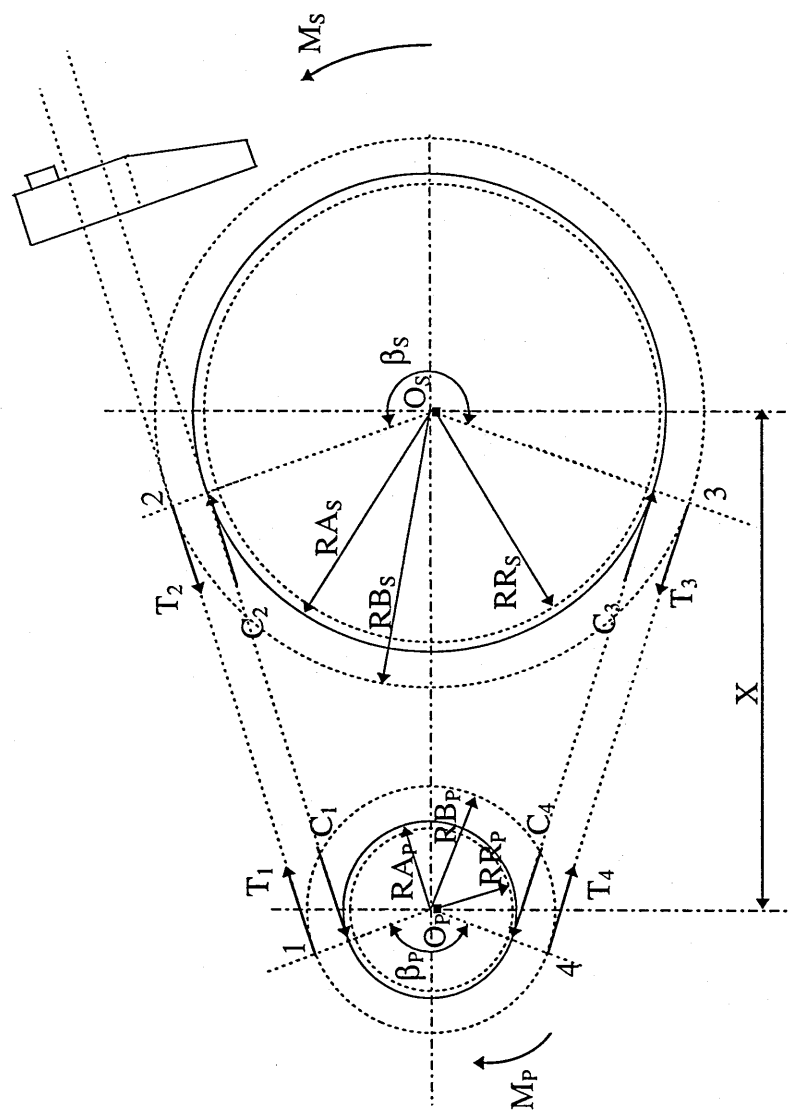


Fig. 5.1 Free body diagram of both primary and secondary pulleys

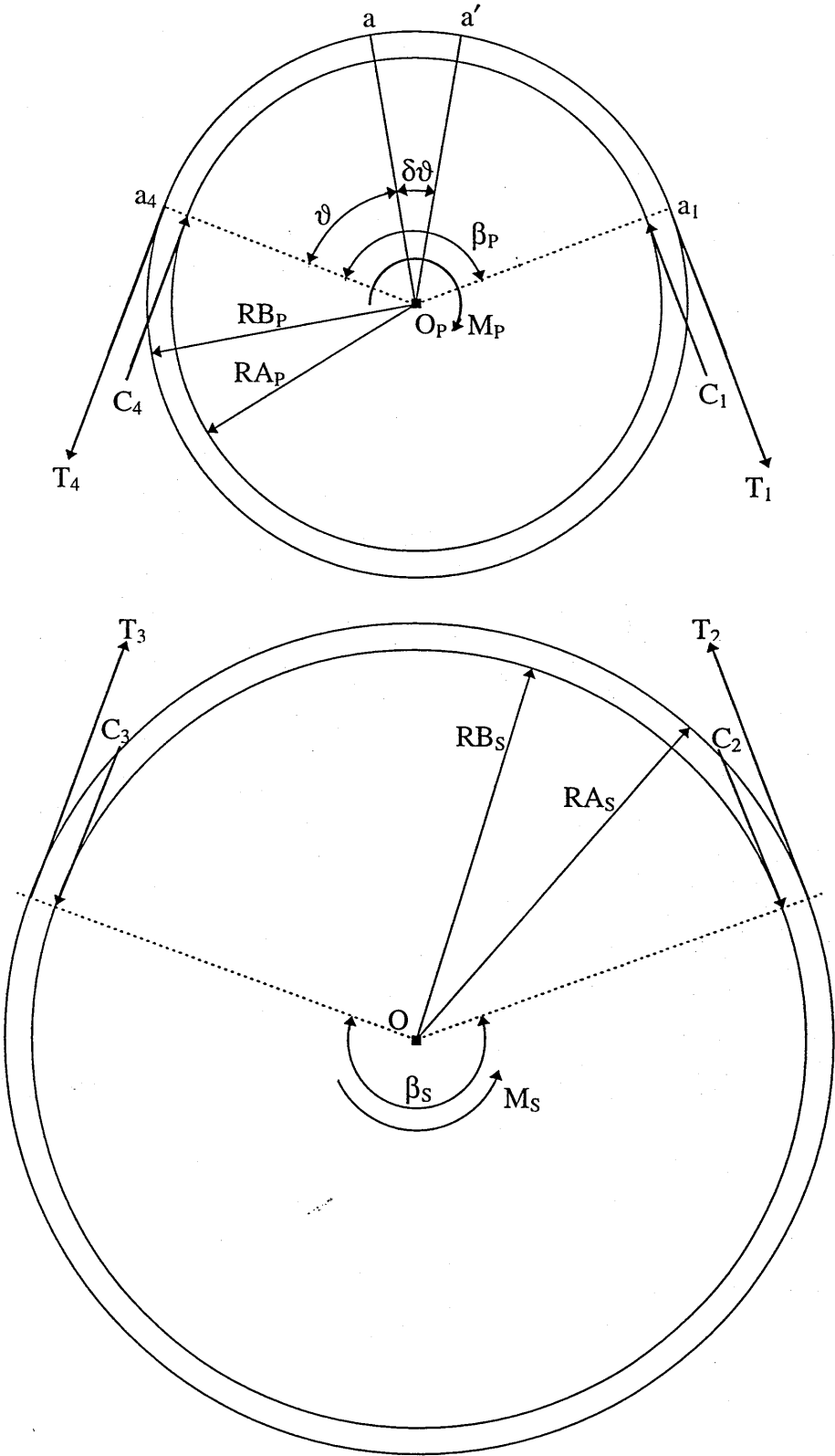


Fig. 5.3 Free body diagram of both primary and secondary pulleys

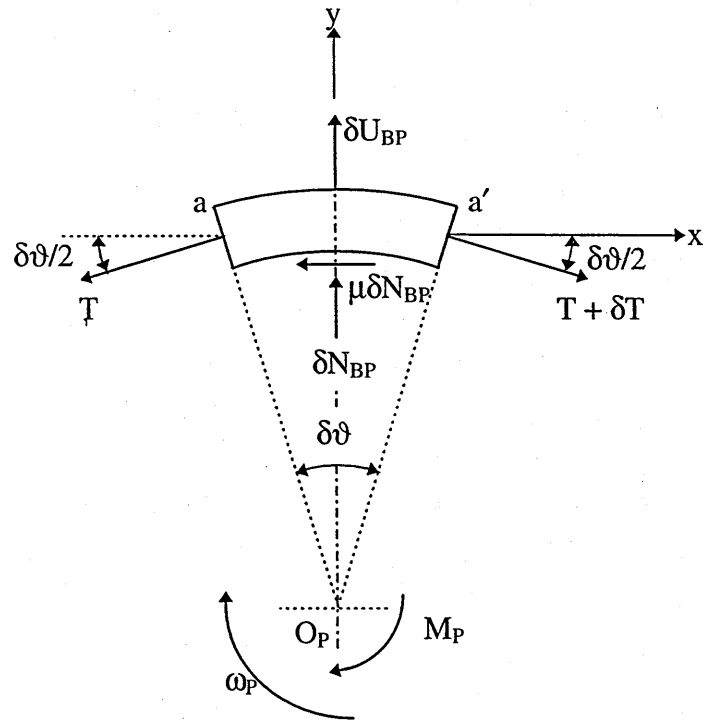


Fig. 5.4 Free body diagram of elemental band for the primary pulley

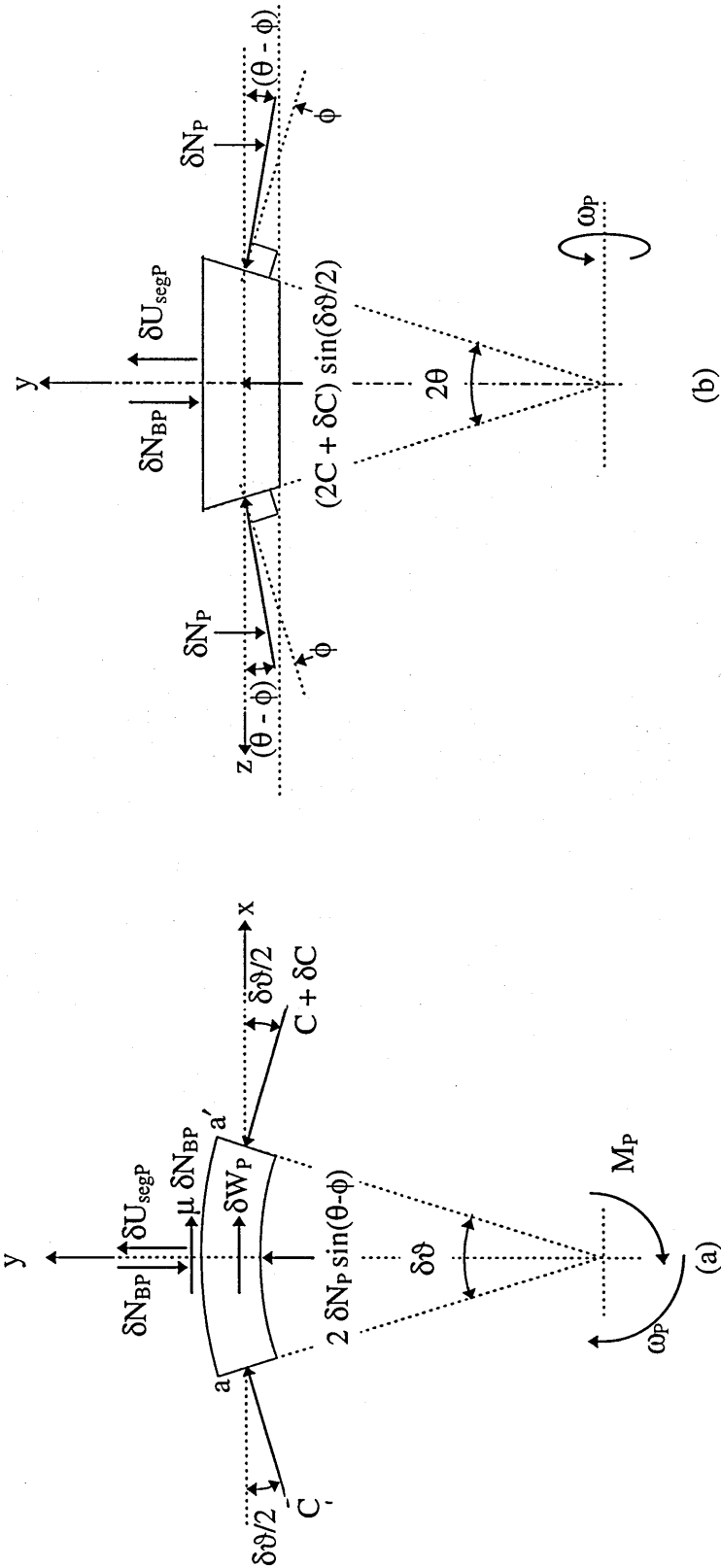


Fig. 5.5 Free body diagram of segment at the primary pulley

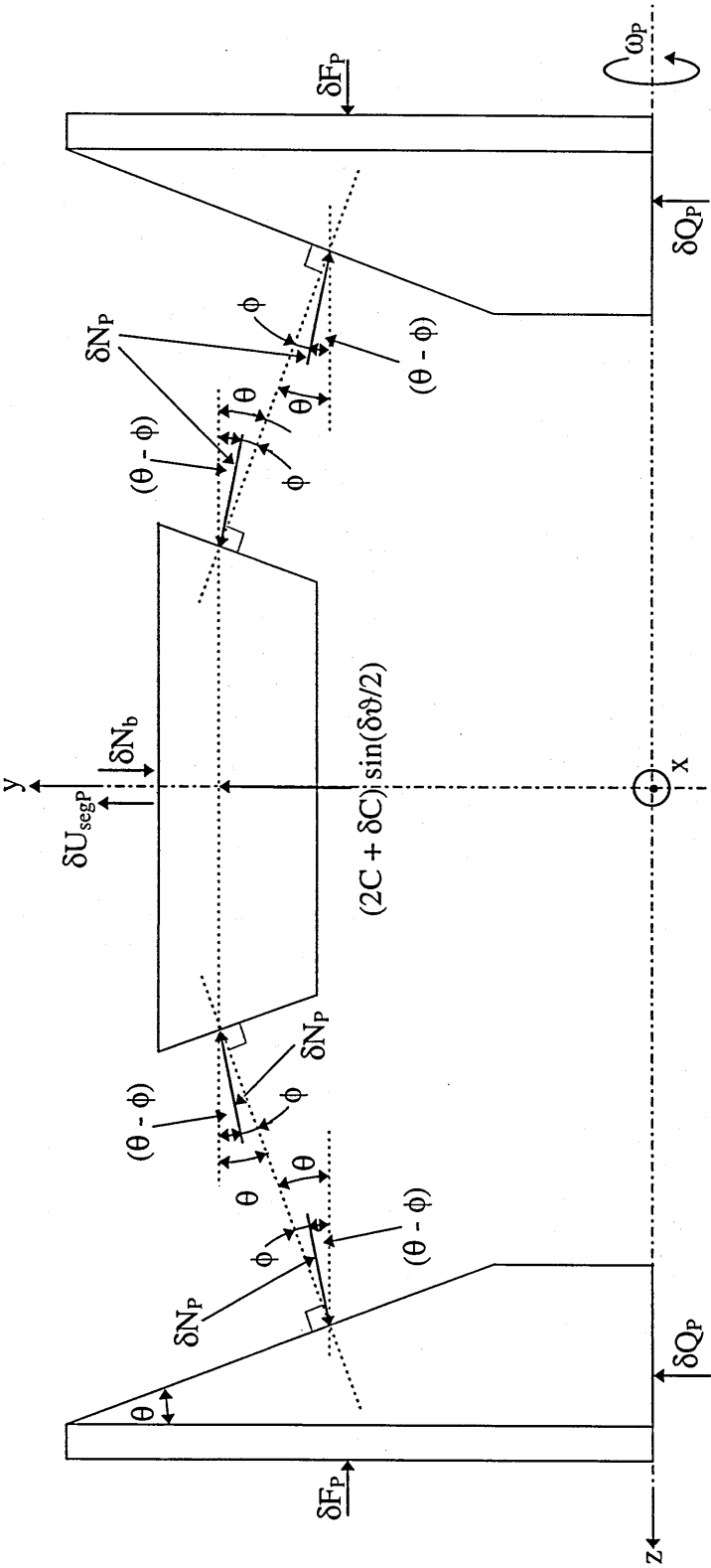
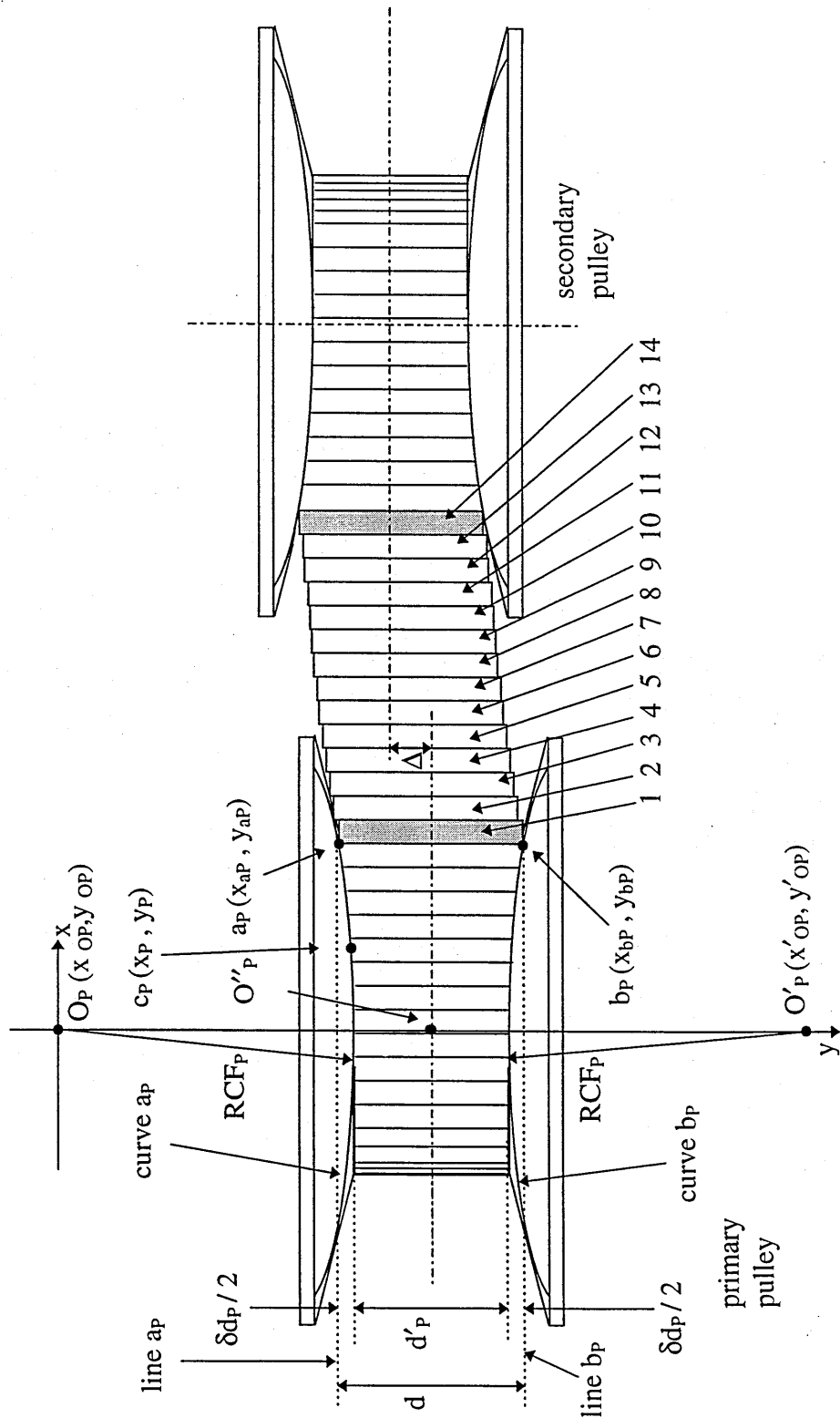


Fig. 5.6 Free body diagram of segment and the primary pulley cones

Fig. 5.7 Schematic diagram of segments position (plan view)



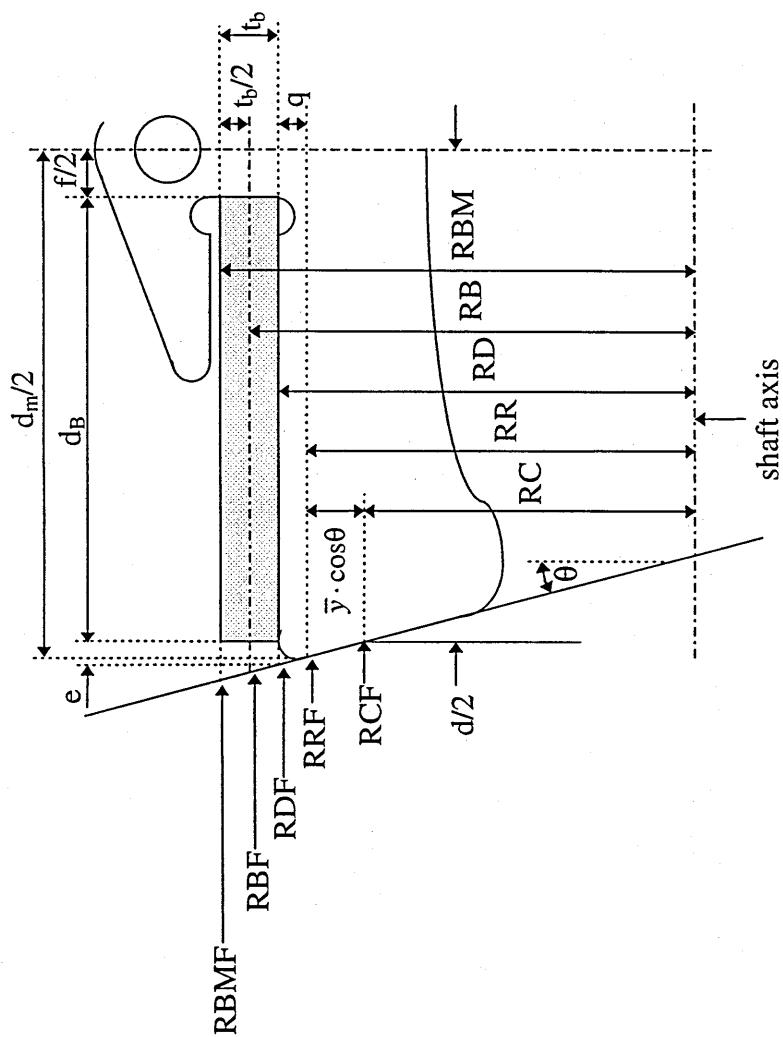
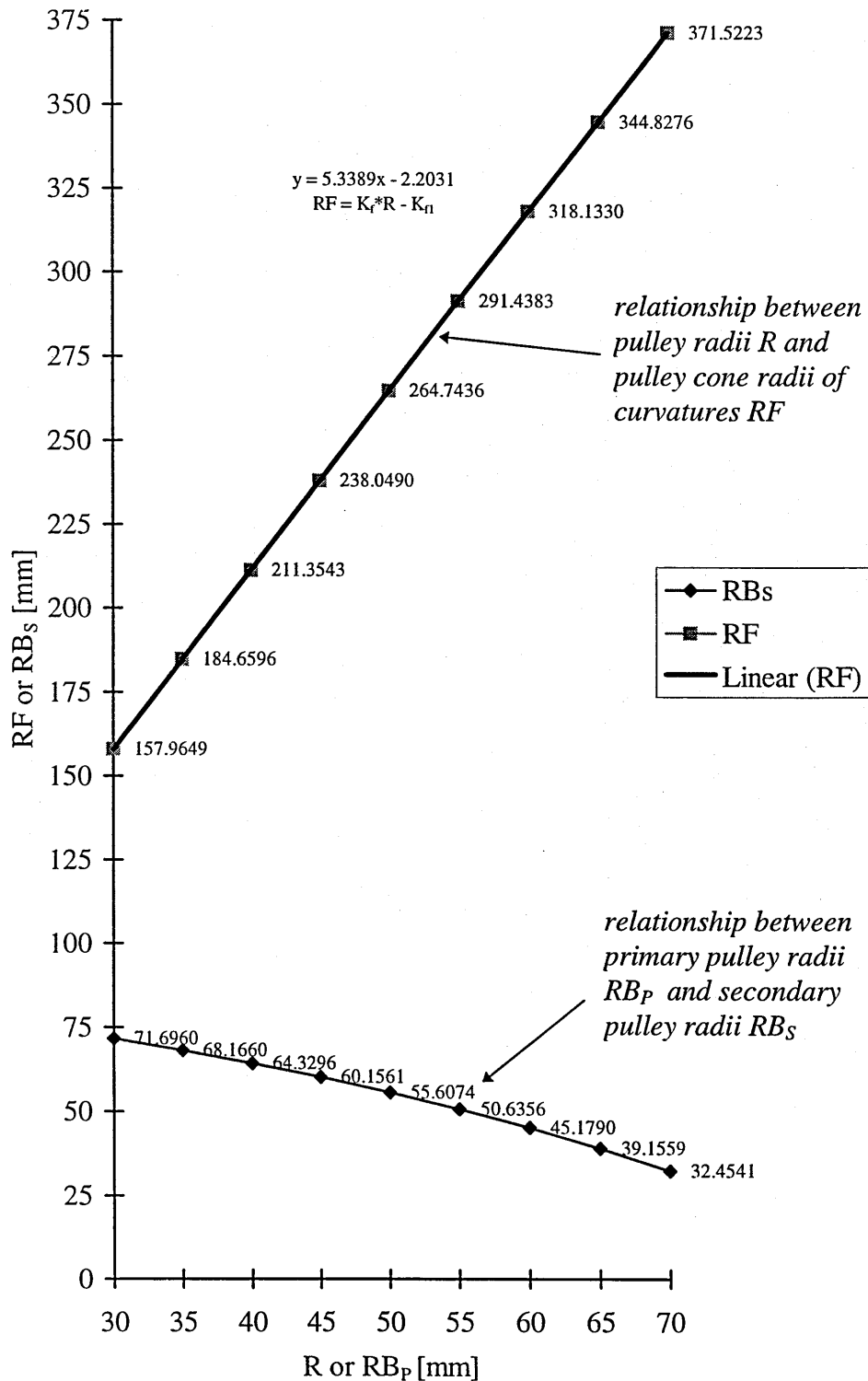


Fig. 5.8 Schematic geometrical definition diagram of the belt

Fig. 5.9 Graphs of primary pulley and secondary pulley radii, and pulley radii and pulley cone radii of curvatures



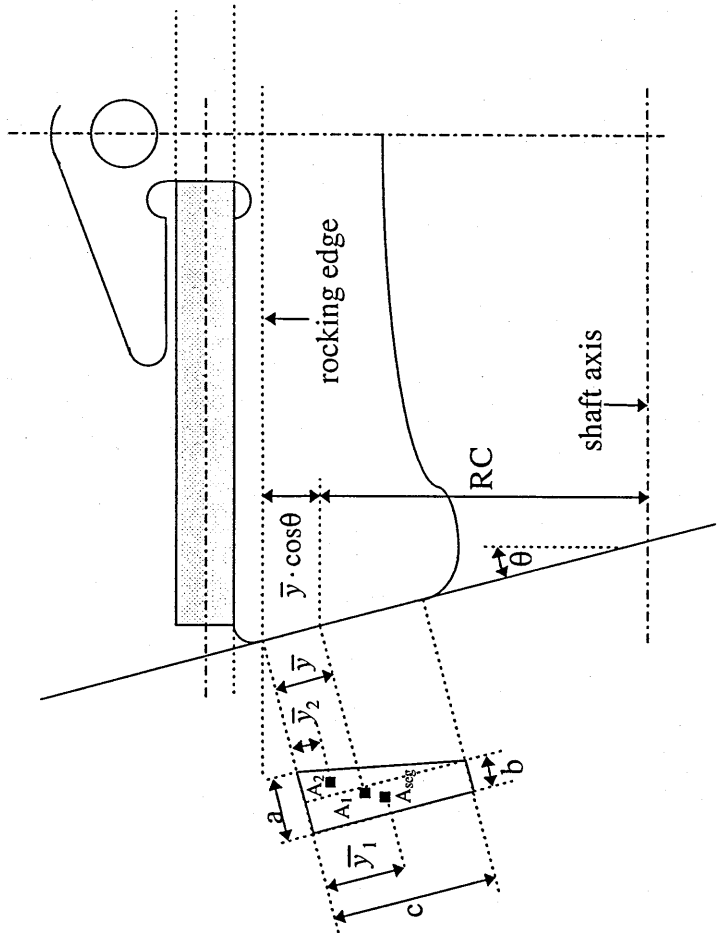


Fig. 5.10 Schematic geometrical definition diagram of the segment side

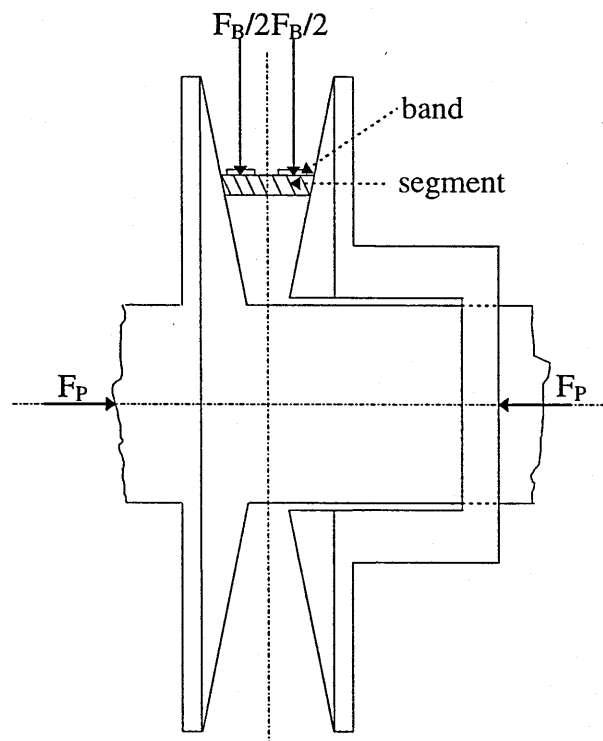


Fig. 5.11 Free body diagram of segments and primary pulley cones

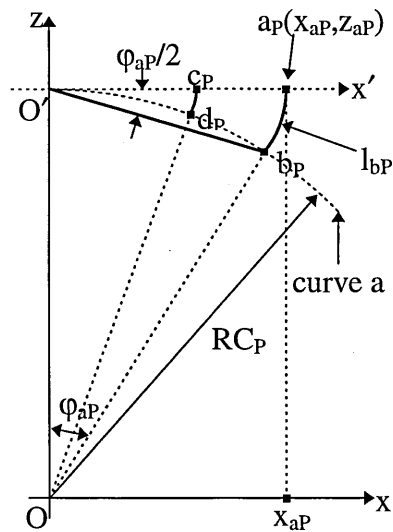


Fig. 5.12 Schematic diagram of belt sliding locus at the primary pulley

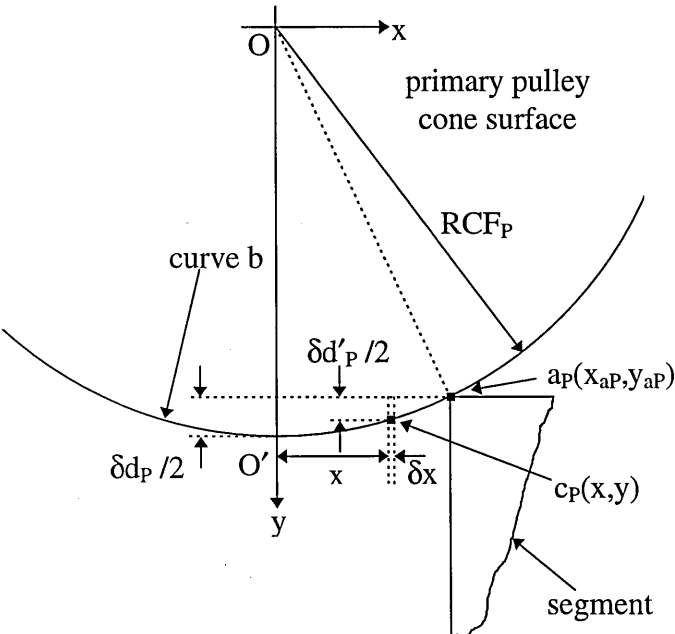


Fig. 5.13 Plan view of Fig. 5.12

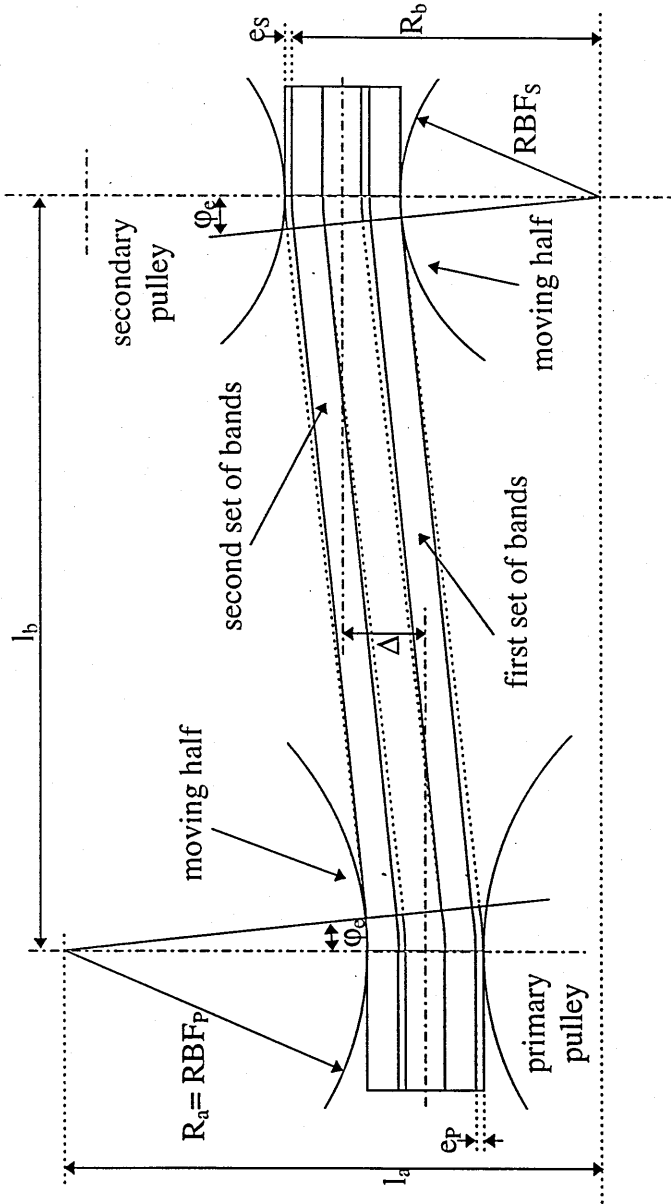


Fig. 5.14 Schematic diagram of the two sets of band free play due to belt misalignment (plan view)

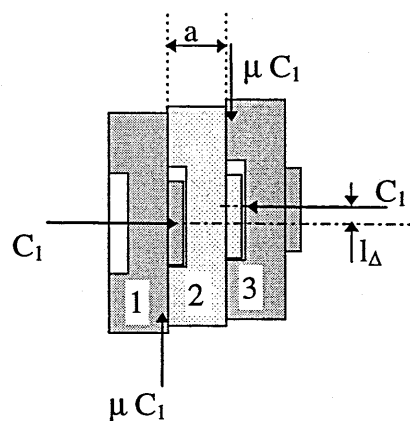


Fig. 5.15 Free body diagram of segment along the compression straight

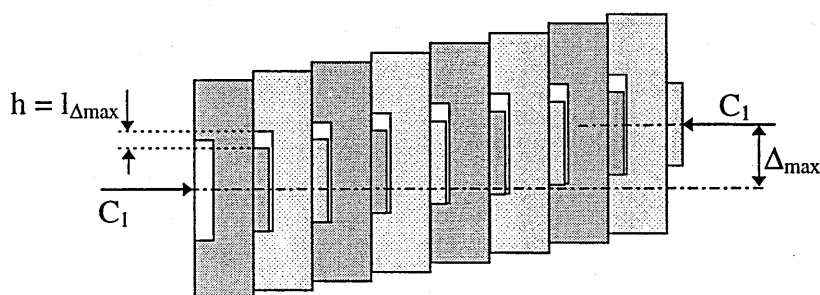
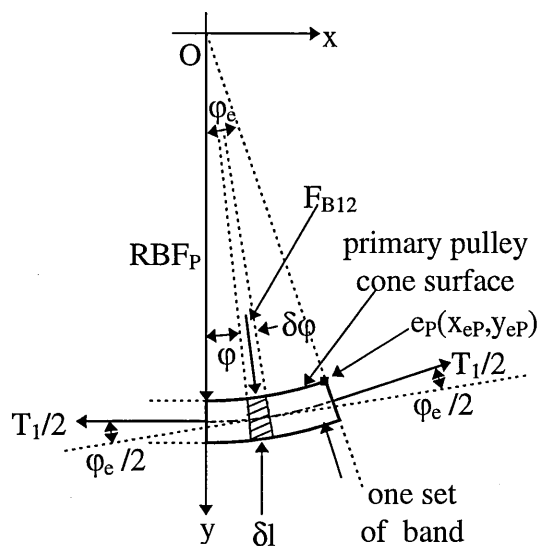
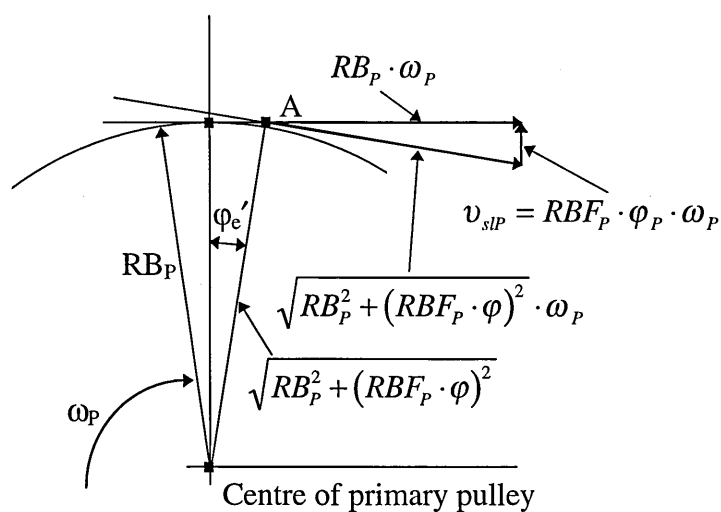


Fig. 5.16 Free body diagram of segment along the compression straight



(a)



(b)

Fig. 5.17 Free body diagram of one set of bands at the primary pulley exit during excessive belt misalignment

R_p [mm]	R_s [mm]	R_F [mm]
30.00	71.6960	157.9649
35.00	68.1660	184.6596
40.00	64.3296	211.3543
45.00	60.1561	238.0490
50.00	55.6074	264.7436
55.00	50.6356	291.4383
60.00	45.1790	318.1330
65.00	39.1559	344.8276
70.00	32.4541	371.5223

Table 5.1 Data for the radii relationship between the primary and secondary pulleys for the pulley cone radii of curvature

CHAPTER 6

SAMPLE CALCULATIONS

6.1 Introduction

This chapter will show some of the necessary calculations required to predict the transmission efficiency using the developed mathematical model. The model in short, will first calculate the belt misalignment values of the transmission, then it will calculate the required geometrical behaviour of the pulleys, segments and bands followed by the bands tension and segments compression along the two straights. From here, the model will then begin to calculate the corresponding power losses and hence the transmission efficiency.

To help the author deal with the vast volume of data involved in this work, a computer programming sequence using Microsoft Excel programming language has been developed to aid in carrying out all the required calculations. Firstly, the raw data of the following,

- i. primary pulley speed [signals/seconds],
- ii. secondary pulley speed [signals/seconds],
- iii. primary pulley pressure [volts],
- iv. secondary pulley pressure [volts],
- v. primary pulley torque [volts] and
- vi. secondary pulley torque [volts],

were captured using a Daqview data acquisition system. The digital electronic signals were transferred into Microsoft Excel spreadsheets format and then converted into their corresponding actual units by substituting their values into the appropriate calibration equations. Details of the calibrations using the Excel curve fitting facilities as shown in the appendixes.

6.2 Constants

The constants used in the above mentioned Microsoft Excel programming sequence are as follows:

	<i>Symbols</i>	<i>Values</i>	<i>Units</i>	<i>Remarks</i>
1	<i>L</i>	604.00	<i>mm</i>	Belt length
2	<i>X</i>	140.00	<i>mm</i>	Pulleys centre distance
3	<i>tb</i>	2.00	<i>mm</i>	Band thickness (0.02 mm. * 10 bands)
4	<i>dm</i>	24.00	<i>mm</i>	Unrestrained segment width at rocking edge
5	<i>d</i>	23.2731	<i>mm</i>	Unrestrained segment width at centre of contact area {= $dm - 2*((h*(a+(2*b)))/(3*(a+b)))*\sin(tet)$ }
6	<i>f</i>	5.00	<i>mm</i>	Segment central column width
7	<i>a</i>	2.00	<i>mm</i>	Segment thickness
8	<i>b</i>	1.50	<i>mm</i>	Segment contact area small width
9	<i>h</i>	4.00	<i>mm</i>	Segment contact area height
10	<i>tet</i>	0.1920	<i>rad</i>	Pulley cone angle { $tet=11*PI()/180$ }
11	<i>yba</i>	1.9048	<i>mm</i>	centre of segment contact area { $yba=((h*(a+(2*b)))/(3*(a+b)))*\cos(tet)$ }; Fig. 5.10, equation (5.5.9)
12	<i>area</i>	6.8714	<i>mm²</i>	Segment contact area { $area=0.5*(a+b)*h*\cos(tet)$ }; Fig. 5.10, equation (5.5.18)
13	<i>E</i>	207000	<i>N/mm²</i>	Young's modulus for steel
14	<i>lne</i>	2.7183		In base e
15	<i>mue</i>	0.1		Coefficient of friction
16	<i>phi</i>	0.0997	<i>rad</i>	Friction angle { $phi=ATAN(mue)$ }
17	<i>dB</i>	8.65	<i>mm</i>	Band width
18	<i>q</i>	1.00	<i>mm</i>	Height above rocking edge to segment shoulder
19	<i>mB</i>	0.2551	<i>kg/m</i>	Mass of bands / metre (2 width length i.e. $8.65*2=17.3$ mm)
20	<i>mS</i>	1.2983	<i>kg/m</i>	Mass of segment / metre
21	<i>Kf</i>	5.3389		Gradient for the pulley cone radii of curvature equation
22	<i>Kfl</i>	2.2031	<i>mm</i>	Constant for the pulley cone radii of curvatures equation
23	<i>Rmp</i>	42.2500	<i>mm</i>	Initially assembled primary pulley radius (belt misalignment = 0, speed ratio 0.7188)

24	<i>Rms</i>	60.1527	<i>mm</i>	Initially assembled secondary pulley radius (belt misalignment = 0, speed ratio 0.7188)
25	<i>Adelta</i>	0.00	<i>mm</i>	Add on belt misalignment

6.3 Sample Calculations

In order to start the calculation one of the following values is required,

- primary pulley radius (equation 3.2.5),
- secondary pulley radius (equation 3.2.6) or
- speed ratio (equation 5.6.17).

Then, with some iterations using equations 3.2.5 and 3.2.6 the corresponding value of α (or alpha) can be determined.

The values of the variables that are required are:

- primary pulley pressure,
- secondary pulley pressure,
- primary pulley torque and
- primary pulley speed.

The primary and secondary pulley pressures are required to deduce their corresponding pulley axial compression forces on the segments, while the other two variables will serve as the input values in order to calculate the output from the secondary pulley.

This section will demonstrate a sample calculation of experiment Expt. 4a31 for speed ratio of 0.5329 with an input speed of approximately 829 *rpm* (86.7711 *rad/s*), using the Excel computer programming sequence that has been developed for this work. An 's' at the end of an equation number denotes calculations for the secondary pulley.

	Eqn. No.	Symbols	Equation	Values	Unit
1		alpha		-0.2216	<i>rad</i>
2	3.2.2	betap	$=PI()-(2*\alpha)$	3.5847	<i>rad</i>
3	3.2.3	betas	$=PI()+(2*\alpha)$	2.6985	<i>rad</i>
4	3.2.5	rbp	$=(L-(X*betas*SIN(\alpha)))-$	65.8679	<i>mm</i>

			$(2 \cdot X \cdot \sqrt{1 - ((\sin(\alpha))^2)}) / (2 \cdot \pi())$		
5	3.2.6	rbs	$= X \cdot \sin(\alpha) + rbp$	35.1039	mm
6	5.6.17	i	$= rbs / rbp$	0.5329	
7	EXPT	wi	EXPT	82.8603	10*rpm
8	EXPT	wo	EXPT	161.2695	10*rpm
9	EXPT	pressp	EXPT	0.2353	V
10	EXPT	presss	EXPT	5.0969	V
11	EXPT	torquep	EXPT	1.2234	V
12	EXPT	torques	EXPT	0.5760	V
13		wp	$= wi \cdot 10 \cdot 2 \cdot \pi() / 60 : \{ \text{to convert rpm to rad/s} \}$	86.7711	rad/s
14	App. 1.1	Mp	$= 10.023 \cdot \text{torquep} + 5$	17.2617	N.m
15	5.6.8	Hp	$= wp \cdot Mp$	1497.8140	W
16	5.6.16	ws	$= wp / i$	162.8144	rad/s
17	5.5.6	rcp	$= rbp - (tb/2) - q - (yba \cdot \cos(tet))$	61.9981	mm
18	5.5.6s	rcs	$= rbs - (tb/2) - q - (yba \cdot \cos(tet))$	31.2342	mm
19	5.5.10	rcfp	$= (Kf \cdot rcp) - Kfl$	328.8007	mm
20	5.5.10s	rcfs	$= (Kf \cdot rcs) - Kfl$	164.5541	mm
21	5.5.7	rrp	$= rbp - (tb/2) - q$	63.8679	mm
22	5.5.7s	rrs	$= rbs - (tb/2) - q$	33.1039	mm
23	5.5.11, 5.8.6	rrfp	$= (Kf \cdot rrp) - Kfl$	338.7832	mm
24	5.5.11s	rrfs	$= (Kf \cdot rrs) - Kfl$	174.5367	mm
25	5.5.8	rdp	$= rbp - (tb/2)$	64.8679	mm
26	5.5.8s	rds	$= rbs - (tb/2)$	34.1039	mm
27	5.5.12	rdfp	$= (Kf \cdot rdp) - Kfl$	344.1222	mm
28	5.5.12s	rdfs	$= (Kf \cdot rds) - Kfl$	179.8756	mm
29	5.5.13	rbfp	$= (Kf \cdot rbp) - Kfl$	349.4611	mm
30	5.5.13s	rbfs	$= (Kf \cdot rbs) - Kfl$	185.2145	mm
31	App. 1.14	Fp	$= 105700 \cdot \text{pressp} - 757.19$	24111.7353	N
32	App. 1.9, 1.12, 1.13	Fs	$= \text{presss} \cdot 3219.8 + 11.651 - (16.961 \cdot 49.3 - 29.217) + 16.961 \cdot (-0.0019 \cdot rbs^2 - 0.2016 \cdot rbs + 63.378) + 29.217$	16560.1311	N

33	5.5.17	Fsegp	$=Fp/(rrp*betap/a)$	210.6319	N/seg
34	5.5.17s	Fsegs	$=Fs/(rrs*betas/a)$	370.7601	N/seg
35	5.5.19	d1p	$=Fsegp*d/(E*area)$	0.0034	mm
36	5.5.19s	d1s	$=Fsegs*d/(E*area)$	0.0061	mm
37	5.5.20	d2p	$=d-d1p$	23.2697	mm
38	5.5.20s	d2s	$=d-d1s$	23.2670	mm
39	5.5.15	yap	$=rcfp-(d1p/2)$	328.7989	mm
40	5.5.1s	yas	$=rcfs-(d1s/2)$	164.5511	mm
41	5.5.22	xap	$=SQRT((rcfp^2)-(yap^2))$	1.0645	mm
42	5.5.22s	xas	$=SQRT((rcfs^2)-(yas^2))$	0.9991	mm
43	5.5.23	Xd	$=X*COS(alpha)-(xap+xas)$	134.5145	mm
44	3.2.11	delta	$=Adelta+ABS(((rbs-Rms)-(Rmp-rbp))*TAN(tet))$	-0.2781	mm
45	5.4.9	aa	$=(2*Fp*TAN(tet-phi)/betap)+(Mp/(mue*(rcp/1000)*betap))+(ms*(wp^2)*(rrp/1000)^2)$	2062.0247	N
46	5.4.10	cc	$=aa*((lne^(mue*betap))-1)$	889.0154	N
47	5.6.14	TD	$=(mb*(wp^2)*(rbp/1000)^2)+((((cc*(rbp/1000))-Mp)/(rbp/1000))/((lne^(mue*betap))-1))$	1462.5114	N
48	5.6.5	TA	$=(lne^(mue*betap))*(TD-(mb*(wp^2)*(rbp/1000)^2))+(mb*(wp^2)*(rbp/1000)^2)$	2089.4616	N
49	5.7.1	psiap	$=2*ASIN(xap/(2*rcp))$	0.0172	rad
50	5.7.1s	psias	$=2*ASIN(xas/(2*rcs))$	0.0320	rad
51	5.7.20	PLslp	$=((4*mue*Fp*wp*psiap/(betap*d1p))*(((rcfp^2)/2)*((ASIN(xap/rcfp))+(xap*(SQRT((rcfp^2)-(xap^2)))/(rcfp^2)))-(xap*yap)))/1000$	1.4224	W
52	5.7.21	PLsls	$=((4*mue*Fs*ws*psias/(betas*d1s))*(((rcfs^2)/2)*((ASIN(xas/rcfs))+(xas*(SQRT((rcfs^2)-(xas^2)))/(rcfs^2)))-(xas*yas)))/1000$	4.2579	W
53	5.7.22	PLsl	$=PLslp+PLsls$	5.6803	W
54	5.8.1	ep	$=((d2p/2)-(db+(f/2)))+(q*TAN(tet))$	0.6792	mm
55	5.8.2	es	$=((d2s/2)-(db+(f/2)))+(q*TAN(tet))$	0.6779	mm
56	5.8.4	la	$=rbfp+d2p+rbfs-delta$	557.6671	mm
57	5.8.5	lb	$=X*COS(alpha)$	136.5781	mm

58	5.5.13, 5.8.6	Ra,rbfp	$= (K_f * r_{bp}) - K_{f1}$	349.4611	mm
59	5.8.7	Rb	$= r_{bfs} + (d2s/2) + (f/2) + db$	207.9981	mm
60	5.8.3	psie	$= IF(OR(ABS(delta) \geq ep, ABS(delta) > es), ATAN(1/(((-1 * la * lb) - SQRT(((la^2) * (lb^2)) - (((la^2) - ((Ra + Rb)^2)) * ((lb^2) - ((Ra + Rb)^2)))))) / (((la^2) - ((Ra + Rb)^2))))), 0)$	0.0000	rad
61	5.8.16	PLsml	$= (2 * \mu_{cc} * ABS(delta) * w_p * r_{bp} / (X_d - a)) / 1000$	2.1330	W
62	5.8.24	PLex1	$= IF(OR(psie < 0, psie = 0), 0, ((\mu_{TA} * w_p * (r_{bfp} / 1000) * (psie^2 / 4)))$	0.0000	W
63	5.8.25	PLen2	$= IF(OR(psie < 0, psie = 0), 0, ((\mu_{TA} * w_s * (r_{bfs} / 1000) * (psie^2 / 4)))$	0.0000	W
64	5.8.26	PLex3	$= IF(OR(psie < 0, psie = 0), 0, ((\mu_{TD} * w_s * (r_{bfs} / 1000) * (psie^2 / 4)))$	0.0000	W
65	5.8.27	PLen4	$= IF(OR(psie < 0, psie = 0), 0, ((\mu_{TD} * w_p * (r_{bfp} / 1000) * (psie^2 / 4)))$	0.0000	W
66	5.8.29	PLexc	$= IF(OR(psie < 0, psie = 0), 0, 2 * (PLex1 + PLen2 + PLex3 + PLen4))$	0.0000	W
67	5.9.1	TPL	$= PLsl + PLsml + PLezc$	7.8133	W
68	5.6.15	Msi	$= M_p * r_{bs} / r_{bp}$	9.1995	N.m
69	5.6.18	Msa	$= M_{si} * (1 - (TPL / H_p))$	9.1515	N.m
70	App. 1.1	Ms	$= 10.023 * torques$	5.7736	N.m
71	5.6.20	MLs%	$= (1 - (M_{sa} / M_{si})) * 100$	0.5216	%
72	5.6.19	Eff	$= M_{sa} * 100 / M_{si}$	99.4784	%
73	3.2.8	deltap0	$= ((R_{mp} - r_{bp}) * TAN(tet))$	-4.5908	mm
74	3.2.9	deltas0	$= (r_{bs} - R_{ms}) * TAN(tet)$	-4.8690	mm
75	3.2.11	delta0	$= deltas0 - deltap0$	-0.2781	mm
76	3.2.8	deltap2	$= ((R_{mp} - r_{bp}) * TAN(tet))$	-4.5908	mm
77	3.2.9	deltas2	$= (r_{bs} - R_{ms}) * TAN(tet) - 2$	-6.8690	mm
78	3.2.11	delta2	$= deltas2 - deltap2$	-2.2781	mm

Item 72 is used in Fig. 8.18 as the calculated transmission efficiency value. Items 73 and 74 are used in Fig. 8.2 to deduce the calculated values of the belt axial movements l_p and l_s , respectively; similarly, items 76 and 77 for Fig. 8.6. Items 75 and 78 are used in Fig. 8.3 and 8.7 respectively, for the calculated belt misalignment values.

CHAPTER 7

EXPERIMENTAL WORK

7.1 Experimental Rig

The test rig was designed and built so that experimental measurements could be made to compare and study the results from the theoretical model put forward by the author. Fig. 7.1 shows the schematic diagram of the system that was designed. Power was supplied to the input or primary pulley shaft by an a.c. motor capable of transferring 2 hp (1.50 kW) at 400 rpm and 9 hp (6.75 kW) at 1750 rpm. A tooth-belt was used to transfer power from the a.c. motor to the input shaft with 1:1 speed ratio. The tooth-belt was used to ensure that there is no slipping taking place between the a.c. motor and the input shaft. At the output shaft or secondary pulley, power is then absorbed by an eddy-current dynamometer capable of absorbing 4 hp (3 kW) at 400 rpm and 50 hp (37.5 kW) at 6000 rpm. These by themselves explain the limitation of this work. Thus, a speed reduction tooth-belt of about 1.71 was used from the output shaft to the dynamometer. This was just to ensure that, the power available from the a.c. motor could be maximised and absorbed by the dynamometer over all the transmission speed ratio range of about 0.5 to 2.5. Input torque and speed were measured by a torque and speed meter mounted between the a.c. motor and the transmission input shaft, while the output torque and speed were measured by another torque and speed meter mounted between the eddy-current dynamometer and the transmission output shaft. The primary and secondary pulley pressures were supplied externally and independently and were measured by using pressure transducers.

The Van Doorne metal pushing V-belt CVT, with both the primary and secondary pulleys system have been disassembled from their original casing and assembled again into a specially designed transmission box. The new transmission box was designed because of the necessity to measure the appropriate parameters such as belt radii and misalignments (section 7.2). Also, the new transmission box has been designed in such a way that it could be dismantled and assembled quickly, so that additional artificial belt misalignment *adelta*, could be introduced for experimental purposes (section 7.6).

In this experimental work, it was necessary to know the relationship between the secondary pulley pressure and its axial compression force on the segments sides. In addition the secondary pulley has a built in spring, therefore its sliding pulley half axial movement also had to be calibrated to incorporate it into the pulley compression force (section 7.4). Where as, for the primary pulley, only its pressure and its compression force on the segments sides needs to be calibrated because it has no built in spring (section 7.5). These had to be carried out because, the theoretical model developed by the author requires the values of both pulley axial compression forces in order to deduce the bands tension and segments compression along the two straights and hence, calculate the power loss due to belt misalignments.

The transmission oil that was supplied from an outside tank, was delivered using an a.c. motor and pump capable of delivering up to 1000 *psi* (68.95 *bar*). Unfortunately, the cover which replaced the transmission oil pump where the external supply pipes for the hydraulic pressures were fitted to, had a small leak. Modification of the adapter was made but it only reduced the leaking. Since time did not permit any further improvement, the author opted to use a method of capturing the leaking oil. The inevitable internal leaking of the transmission oil between the adjustable pulley halves and fixed shaft further reduced the hydraulic supply pressure. Due to these, the pump could only delivered up to a constant maximum of about 300 *psi* (20.68 *bar*) constantly. Thus, a constant secondary pulley pressure of about 280 *psi* (19.31 *bar*) was chosen for this work due to this limitation. This compares with the line pressure of 313 *psi* (22 *bar*) given in the transmission specification. As only a relatively small proportion of the maximum torque was applied to the primary pulley this small loss of pressure was considered acceptable.

7.1.1 Instrumentation

Only the pulley torque, speed and pressure were measured with the aid of electrical apparatus. All other measurements such as belt radii and belt misalignments were measured using mechanical and optical instrumentation.

7.1.2 Torque Measurements

Input and output torque were measured using torque meters with the aid of specially designed circuit board by Cranfield University's electronic engineer. Both torque meters were calibrated manually using a cantilever torque arm and a set of dead weights. The specially designed electric circuit board allowed the gain to be set up accordingly so that, both the torque meters had almost the same calibration curve (Appendix 1.1) but with different zero offset. The secondary torque meter had zero offset while the primary one had -0.5 V .

7.1.3 Speed Measurements

Primary and secondary pulley speeds were measured using the same torque and speed meters that were used to measure the primary and secondary pulley torque. Six signals per revolution from the torque and speed meters were measured using electronic probes and were then converted into revolutions per minute using Electronic Digital Speed Counters. Both the primary and secondary pulleys speeds were checked using a hand held tachometer and were found to be within a few revolutions per minute.

7.1.4 Pressure Measurements

Pressure measurements for both primary and secondary pulleys were measured using pressure transducers. The pressure transducers were calibrated using dead weight pressure calibration equipment. Appendix 1.2 and 1.3 show the calibration curves for the primary and secondary pulleys pressure transducers, respectively.

7.2 Belt Misalignment Measurements

7.2.1 Introduction

The new transmission box has been designed with viewing windows (using clear perspex with two small apertures) for viewing the position using the optical instrumentation. The optical instrumentation (Fig. 7.2), consists of one adjustable microscope that is capable of focusing up to approximately 150 mm , a linear micro-positioner that is capable of measuring up to 50.00 mm linearly and a rotational micro-positioner that is capable of measuring a full $360^{\circ}00''$ turn. Fig. 7.3 shows, how the belt radii measurement were taken and measured. Since the transmission belt

misalignment values are the same at their corresponding pulley radii, calibrations exercise were carried out to produce a relationship of the belt axial movements along their respective shafts and hence, deducing the actual belt misalignment values.

7.2.2 Calibrating Belt Radii

First of all, the optical instrumentation was mounted on the transmission box in a position such that, the rotational micro-positioner and the adjustable microscope axes of rotation coincide and are parallel to both the primary and secondary pulley shafts (Fig. 7.2). To ensure mounting accuracy of the optical instrumentation on the transmission box, the mounting surface was machined relative to the shaft centres in the University workshop. To measure the belt radii, the eye-piece horizontal cross-wire was positioned so that it is in line with the top surface of the band (Fig. 7.4). Then the angle ϑ_{mP} and the actual belt radii RB_p and RB_s were then calibrated. The transmission was originally set at underdrive position with both pulley pressures equal to the running experimental condition; while the output braking torque of the eddy-current dynamometer was set at about 15.5 N.m (that is, approximately mid-way running experimental condition). These were to ensure that, the belt radii were calibrated at as similar conditions as possible to the experimental running conditions. The transmission was allowed to run for a few minutes at the required speed ratio. It was then stop naturally by switching off the a.c. driving motor for the appropriate measurements to be taken. The actual belt radii were measured using a digital depth gauge vernier. Appendix 1.4 shows the calibration curves between the optical measuring angle ϑ_{mP} (*tetamP*) and the actual belt radii RB_p and RB_s respectively.

7.2.3 Calibrating Belt Misalignment

The transmission (with similar braking torque setting as in the previous section) was initially set to run at about the required condition for sometime. Then it was stopped for the belt misalignment calibration measurements to be carried out. First, the eye-piece horizontal cross-wire was positioned so that it was in line with the top surface of the band and its vertical cross-wire was in line with the side of the segment hats (Fig. 7.4), at the primary pulley. The linear micro-positioner or the belt axial movement distance of the primary pulley l_p (Fig. 7.2) was then taken as a reference position.

Then, the microscope tube was rotated to the secondary pulley and the same procedure was repeated again to the same set of bands to measure the belt axial movement of the secondary pulley l_s (Fig. 7.2). Appendix 1.5 and 1.6 show the calibration curves against the optical measuring angle ϑ_{mP} (*tetamP*) and speed ratios for the belt axial movement respectively, for $\delta = 0.00 \text{ mm}$; while Appendix 1.7 and 1.8 show the calibration curves against the optical measuring angle ϑ_{mP} (*tetamP*) and speed ratios for the belt axial movement respectively, for $\delta = 2.00 \text{ mm}$.

7.2.4 Belt Misalignment Measurements

The magnitude of the belt misalignment δ was then deduced from the difference between the two belt axial movement values along their respective shafts as calibrated in the previous section (section 8.2). That is,

$$\delta = l_s - l_p \quad (7.1)$$

7.3 Experiment 1:- To Calibrate The Secondary Pulley Axial Spring Force Against Its Axial Displacement

7.3.1 Introduction

The author unfortunately has no information on the internal spring and the geometrical dimension of the adjustable halves of the secondary pulley hydraulic piston. Since the theoretical model which the author has developed, requires the values of the axial compression forces exerted by the pulley cones on the segments, and it is difficult to measure these forces directly to a reasonable accuracy, thus this exercise was carried out to aid the author to obtain the required information. This experiment was devised to calibrate the secondary pulley axial spring force against its axial displacement. The main advantage of calibrating it this way was that, any frictional losses within the

hydraulic piston system will be taken into consideration. Therefore, the secondary pulley radii had to be calibrated first against its groove expansion *exps*, before calibrating against its internal spring. Appendix 1.9 shows the results of the calibration. Secondly, a load cell that was used through out the calibration exercises was calibrated against tension and compression load. The load cell is capable of taking a maximum tension load of up to 40 *kN* in tension and 20 *kN* in compression. Appendix 1.10 shows the load cell tension calibration curve while, Appendix 1.11 shows the load cell compression calibration curve.

7.3.2 Calibrating Procedure

The whole of the secondary pulley system including its shafts was dismantled from the transmission box. By using a specially designed test rig, it was then held and secured as if it was held in the transmission box. That is, when the spring is being compressed or the pulley groove being expanded accordingly by the test rig, the fixed half pulley shaft will be in tension, thus preventing both of its supporting bearings from being damaged. During the whole process of calibration, the pulley pressure was reduced to zero.

7.4 Experiment 2:- To Calibrate The Secondary Pulley Axial Compression Force Against Its Hydraulic Pressure

This calibration had to be carried out because, the relationship data between the secondary pulley hydraulic pressure and the axial displacement of its cones, together with the data obtained from Experiment 1 (section 7.3), needed to be combined in order to obtain the actual axial compression force of the secondary pulley cones against the pulley hydraulic pressure. Thus, by carrying out the appropriate calculations, the secondary pulley hydraulic pressure and its axial compression force was established which take into consideration the piston axial spring force as well. This was done, by externally pumping in the transmission oil into the piston through a specially made adapter. Using the same test rig as in Experiment 1, the pulley cones were first expanded to a fixed distance within the allowable limit. For this particular case, the pulley groove was expanded to 49.30 *mm* (maximum limit by the manufacturer is 52 to 53 *mm*). The secondary pulley pressure (measured using the

secondary pressure transducer) was then calibrated against the load cell (in tension, Appendix 1.10) by increasing the pulley pressure accordingly. Thus, to deduce its actual clamping force, pulley groove expansion also had to be calibrated against its belt radii. Appendix 1.12 shows the results of the calibration curve of the load cell against the secondary pulley pressure transducer; while Appendix 1.13 shows the calibration curve for the pulley groove expansion against its belt radii.

7.5 Experiment 3:- To Calibrate The Primary Pulley Axial Compression Force Against Its Hydraulic Pressure

As in the previous experiments, also due to the lack of geometrical information on the primary pulley hydraulic piston, its hydraulic pressure needed also to be calibrated against its axial compression force. The calibration process was similar to that of the previous experiments. However, since the primary pulley does not have any internal spring, procedure as in Experiment 1 was omitted and thus, the pulley pressure could be calibrated directly using the same procedure as that of Experiment 2, against the load cell (in compression, Appendix 1.11) by using the same test rig. Appendix 1.14 shows the calibrated results of this experiment.

7.6 Experiment 4:- To Study The Belt Misalignment Effects On The Transmission Efficiency

7.6.1 Introduction

Basically this is the bulk of the work of this project. It was carried out so that results from the theoretical model developed by the author could be analysed and compared with the experimental values. The original geometrical set up of the Van Doorne metal pushing V-belt CVT used in this project was not changed. From the study made so far (section 3.2) and from the design of the CVT itself, it is possible to introduce spacers on the secondary pulley shaft, to introduce additional misalignment. For this experiment, a spacer with a thickness of $\delta = 2.00 \text{ mm}$ was made as the increment for the additional belt misalignment. The increment however had a limit (section 5.8.2), thus for this particular work, the additional belt misalignment δ was limited to 2.00 mm only. To achieve this, a specially designed transmission box was built. Two new specially designed couplings had to be manufactured to support

both the primary and secondary pulley shafts in the new transmission box. These couplings were machined to suit the shafts by the University workshop.

For $\delta = 0.00 \text{ mm}$ it means that, both the primary and secondary pulleys were mounted in the new transmission box at their original set up position as they were originally assembled in their original gear box casing. Where as for $\delta = 2.00 \text{ mm}$ it means that, an additional spacer of 2.00 mm was added along the secondary pulley shaft between the fixed pulley half and its end support bearings, and at the same time a 2.00 mm spacer was also removed between the adjustable pulley half and its end support bearings; thus offsetting the pulley or the author would say, increasing the transmission belt misalignment by 2.00 mm . This was done so that, a direct comparison of the transmission performance due to belt misalignment could be carried out and studied.

7.6.2 Experimental Process

The experiment was started with the original Van Doorne metal pushing V-belt set up as it was initially assembled in the original transmission box. That is, without any additional spacers added to increase its belt misalignment. This then acted as a reference for the rest of the experiments. The measured and variable parameters required for these experiments were:-

- i. θ - optical measuring angle, θ_{mp} ; to calculate the corresponding belt radii R_{BP} and R_{BS} .
- ii. δ - additional belt misalignment spacer; to calculate the corresponding belt misalignment, Δ
- iii. ω_P - primary pulley speed; to calculate the corresponding pulley speed, ω_p .
- iv. ω_S - secondary pulley speed; to calculate the corresponding pulley speed, ω_s .
- v. P_P - primary pulley pressure; to calculate the value of the corresponding pulley axial compression force, F_P .
- vi. P_S - secondary pulley pressure; to calculate the corresponding pulley axial compression force, F_S .

- vii. *torqueP*- primary pulley or input torque, M_P .
- viii *torqueS* - - secondary pulley or output torque, M_S .

All the experiments were carried out at constant secondary pulley pressure of about 280 *psi* (19.31 *bar*). Actual line pressure of about 300 *psi* (20.68 *bar*) as stated in the engine and transmission manual could not be achieved due to system limitations as mentioned earlier. However, 280 *psi* (19.31 *bar*) should be adequate because of the maximum experimental torque for this work is less than 40 *N.m*, which is about half of the actual engine torque. There were twenty experiments altogether that were carried out in this work. These experiments were then divided into ten sets. Each set contained two different experiments, Expt. 4a and 4b. The only difference between these two experiments was the additional artificial belt misalignment values *adelta*, all other parameters were maintained at the same values. Expt. 4a was carried out at *adelta* = 0.00 *mm*, while Expt. 4b at *adelta* = 2.00 *mm*. Summary of all the experimental conditions are shown in Table 7.1.

For convenience purposes, all the experiments with *adelta* = 0.00 *mm* were carried out first. Once all these experiments have been completed, the transmission box was dismantled for the extra additional spacer to be fitted in, that is with *adelta* = 2.00 *mm*. Then, all the experiments with *adelta* = 2.00 *mm* were carried out.

Chapter 8, shows all the relevant experimental results of the experiments that were carried out in this work.

7.7 Data Recording

All electronic data were recorded using a Daq Data Acquisition System produced by,
 IOtech, Inc.
 25971 Cannon Road
 Cleveland, OH 44146
 USA

From the Daq Data Acquisition Systems, 200 samples of data were recorded per second for a period of 10 seconds. That is, a total of 2000 samples of data were recorded per one test run. This was to average out any mechanical and electrical noise contributed from the test rig and to make sure that all the relevant set up conditions were recorded accordingly. These data were then transferred into 'Excel' spreadsheets format where, their average values were then calculated and processed using 'Excel Macro' programming facilities.

7.8 Data Recording Procedures

- i. All electrical equipment were switched on about an hour prior to every experiment to allow them to warm up and stabilise.
- ii. All hydraulic valves leading to the transmission were fully opened to reduce wear during start up condition.
- iii. Breaking torque of the eddy-current dynamometer was set accordingly.
- iv. Both hydraulic pumps were switched on.
- v. Electric motor was switched on with its minimum speed setting.
- vi. The system were allowed to run for a few minutes for as warming up exercise.
- vii. Data acquisition were set accordingly to record all the six parameters per one test run. That is, the input and output speed, input and output torque and input and output pulley pressures.
- viii. Optical instrumentation were set accordingly.
- ix. Input speed, braking torque, input and output pulley pressures were adjusted accordingly.
- x. The data acquisition system was finally checked for data recording and all parameters were fine tuned accordingly.
- xi. The data acquisition were then switched on and simultaneously recorded all parameters being monitored and controlled accordingly within the 10 seconds duration per one test run. Note, this step however, needed to be repeated several times before getting the satisfactory results that was required due to the difficulty of maintaining the required steady-state condition.

7.9 Figures

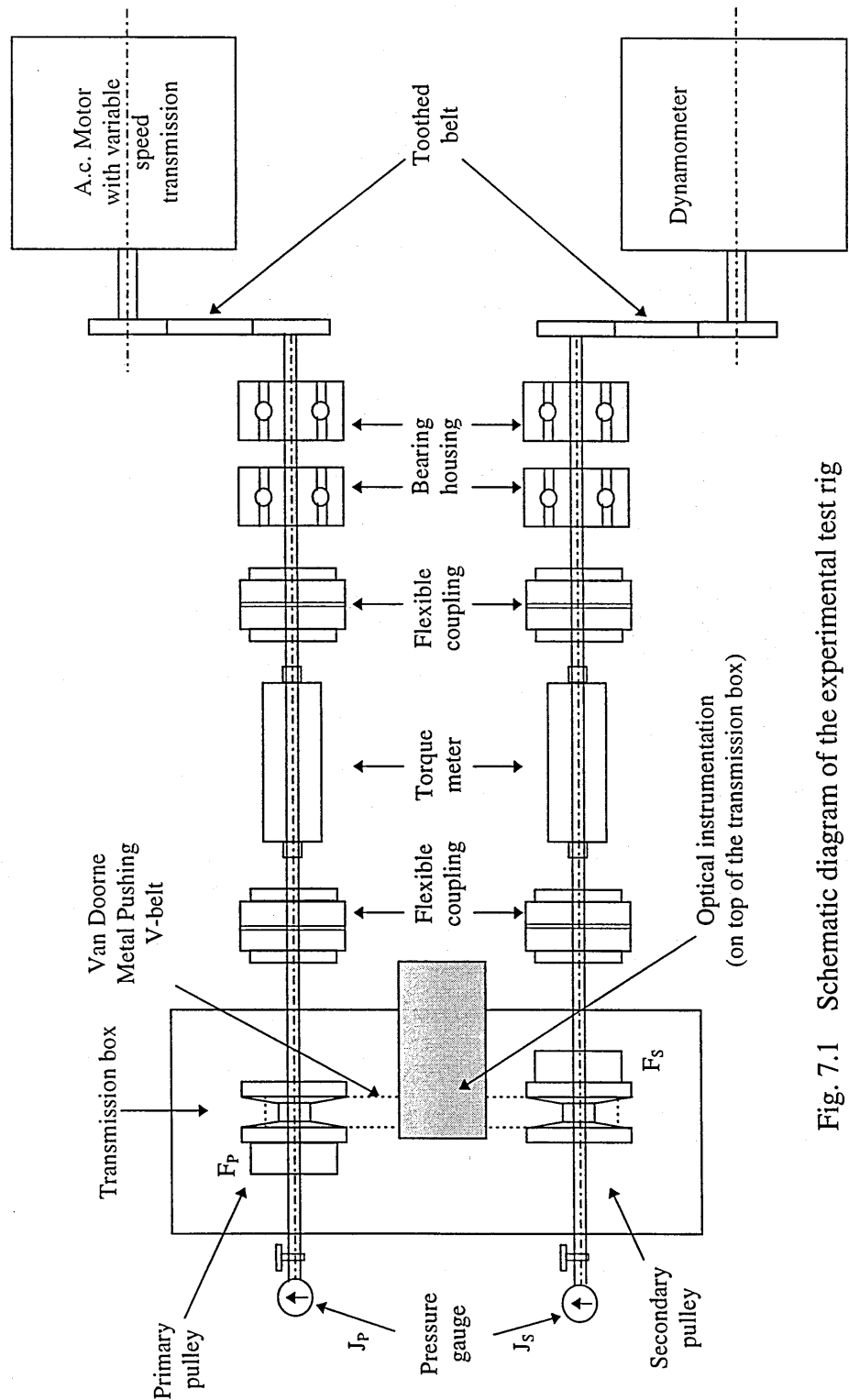
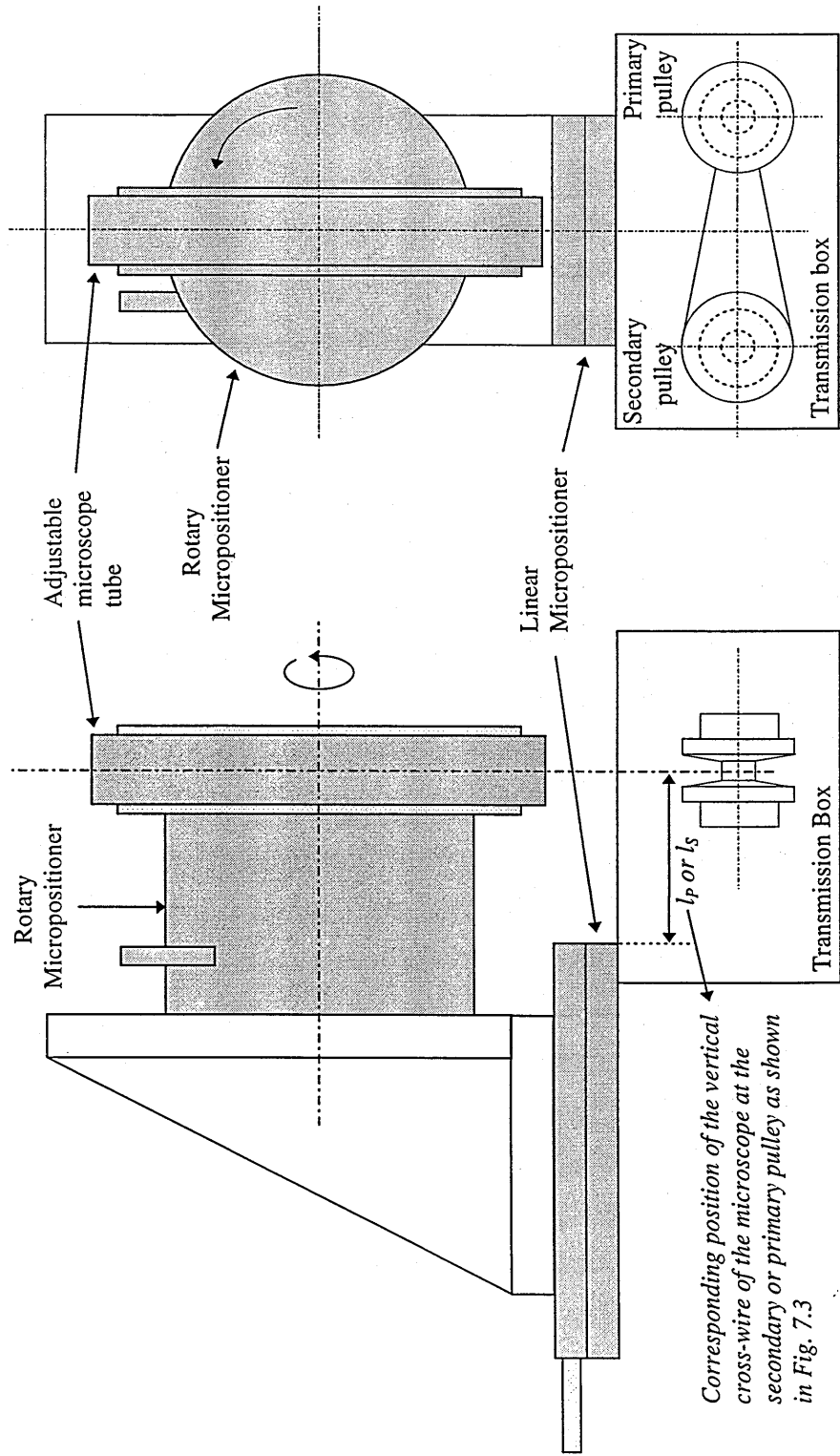


Fig. 7.1 Schematic diagram of the experimental test rig

Fig. 7.2 Optical instrumentation set up



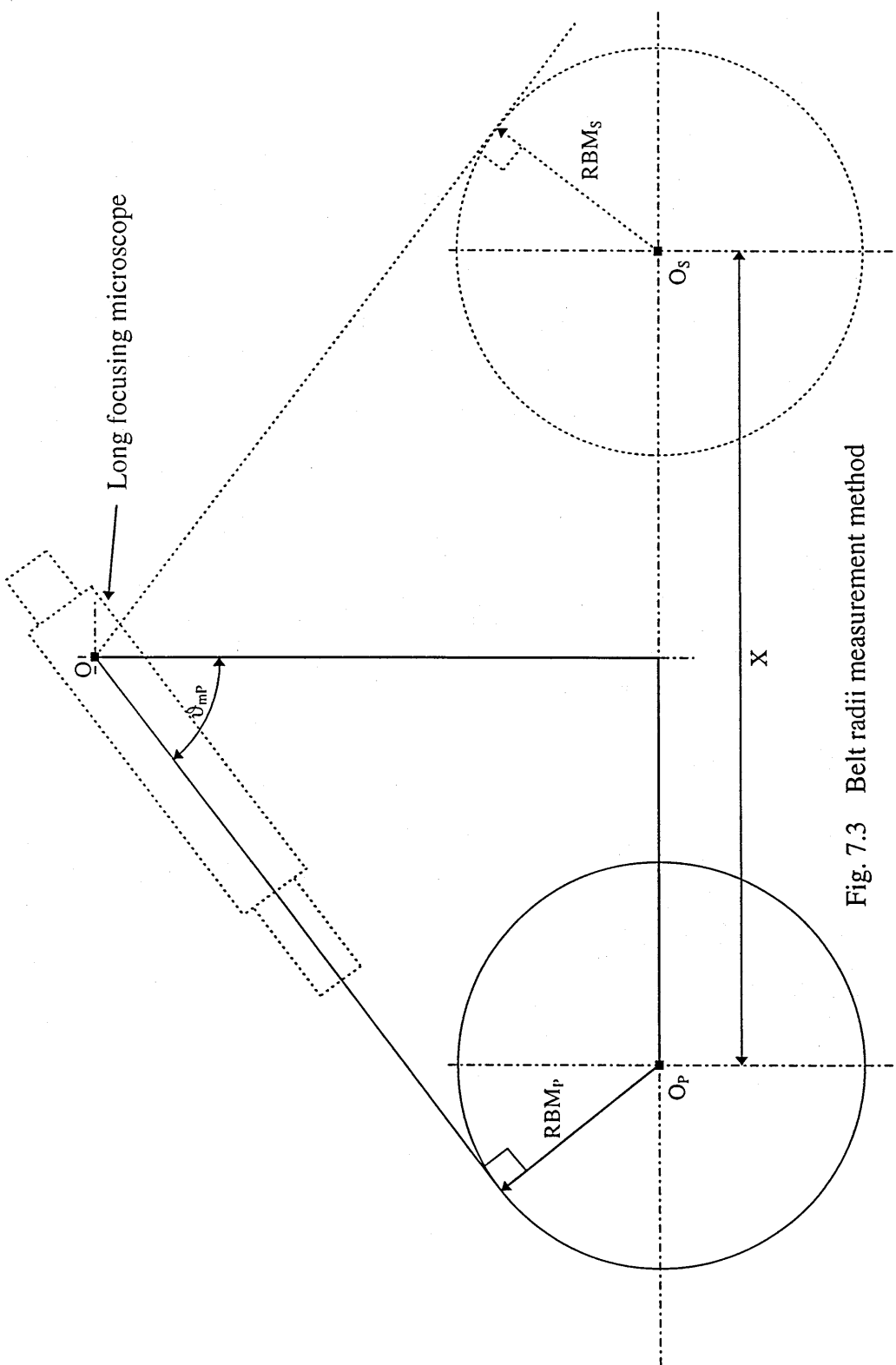


Fig. 7.3 Belt radii measurement method

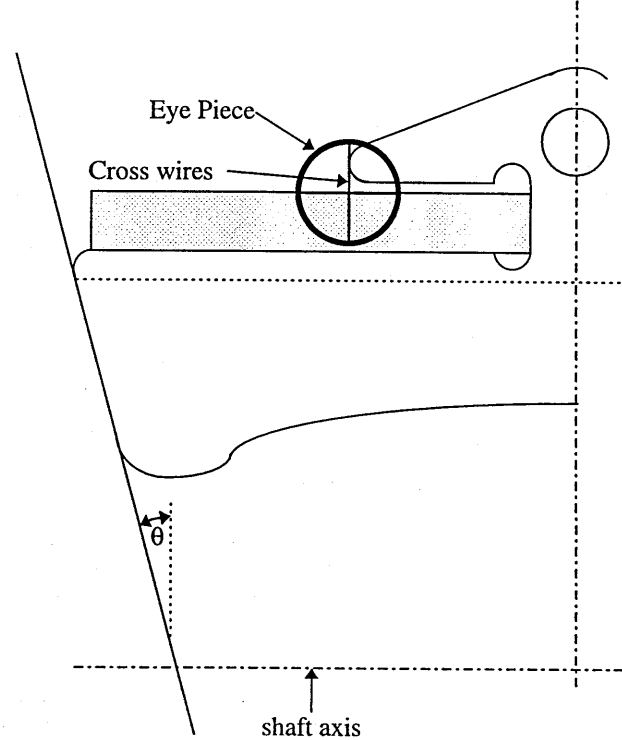
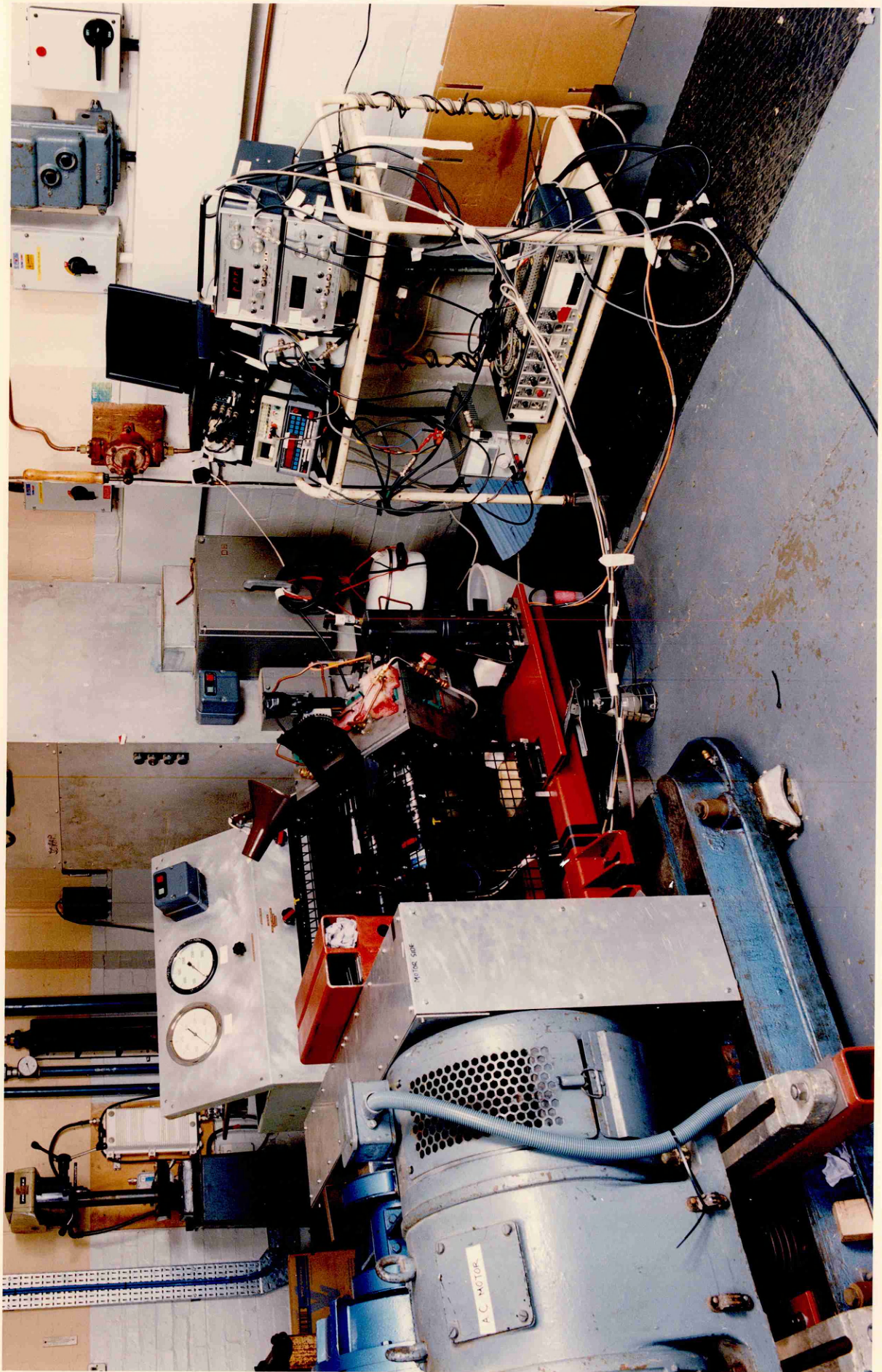


Fig. 7.4 Optical cross-wires position on belt

	Expt.		ωP [rpm]	τP [rpm]	τS [N.m]	RB_P [N.m]
	$\delta = 0.00\text{ mm}$	$\delta = 2.00\text{ mm}$				
1	4a11	4b11	625 - 1750	Constant	Variable	Min.
2	4a12	4b12	625 - 1750	Constant	Variable	Max.
3	4a21	4b21	625 - 1750	Variable	Constant	Min.
4	4a22	4b22	625 - 1750	Variable	Constant	Max.
5	4a31	4b31	825	Constant	Variable	Min. - Max.
6	4a32	4b32	1250	Constant	Variable	Min. - Max.
7	4a33	4b33	1750	Constant	Variable	Min. - Max.
8	4a41	4b41	825	Variable	Constant	Min. - Max.
9	4a42	4b42	1250	Variable	Constant	Min. - Max.
10	4a43	4b43	1750	Variable	Constant	Min. - Max.

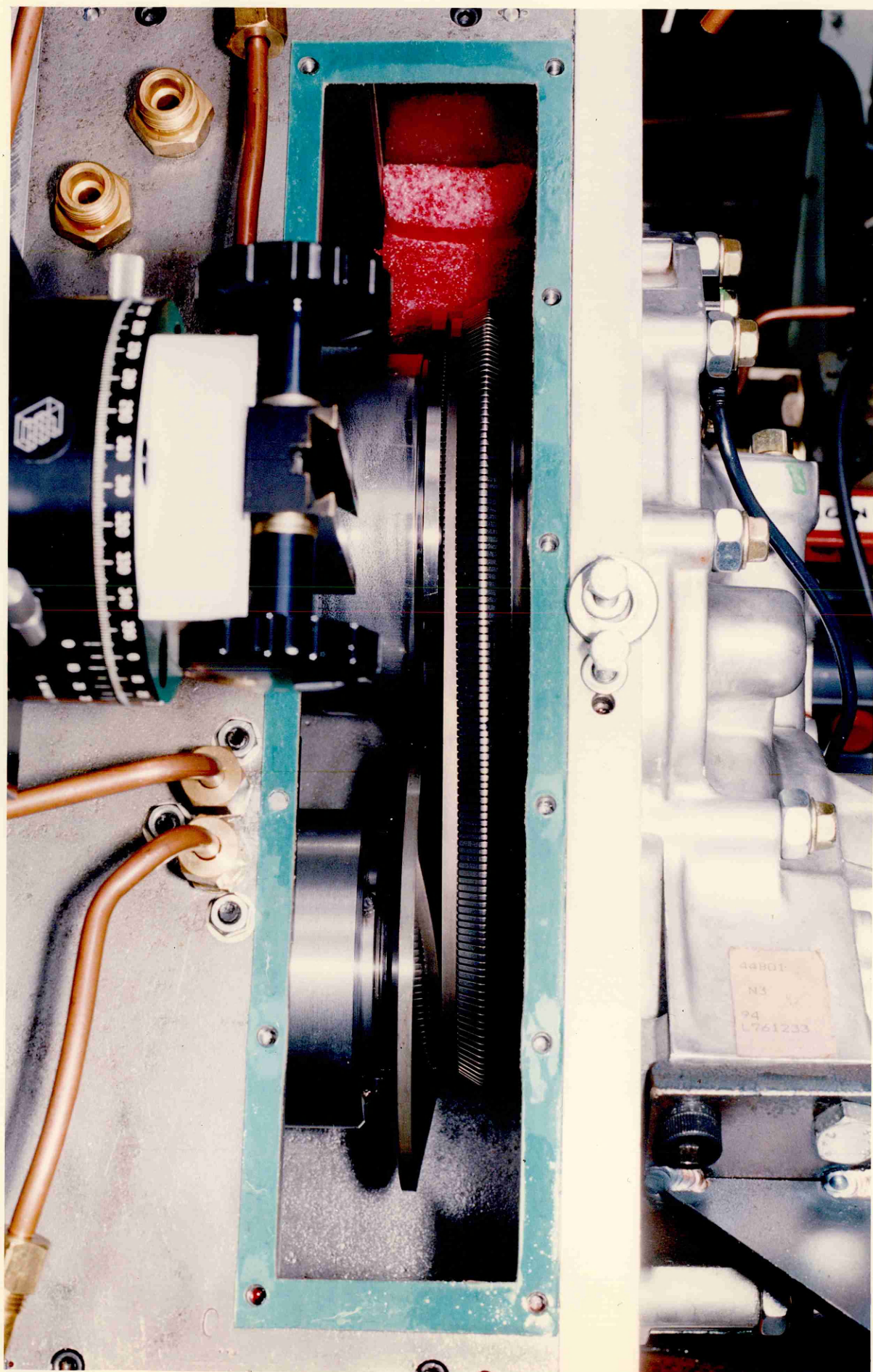
Table 7.1 Summary of Experiment 4

($\delta = 0.00\text{ mm}$. for Expt. 4a and $\delta = 2.00\text{ mm}$. for Expt. 4b.)

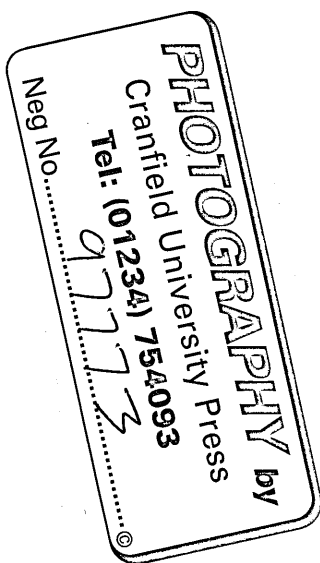


GENERAL VIEW OF TEST RIG

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METAL PUSHING BELT WITH MISALIGNMENT



PHOTOGRAPHY by

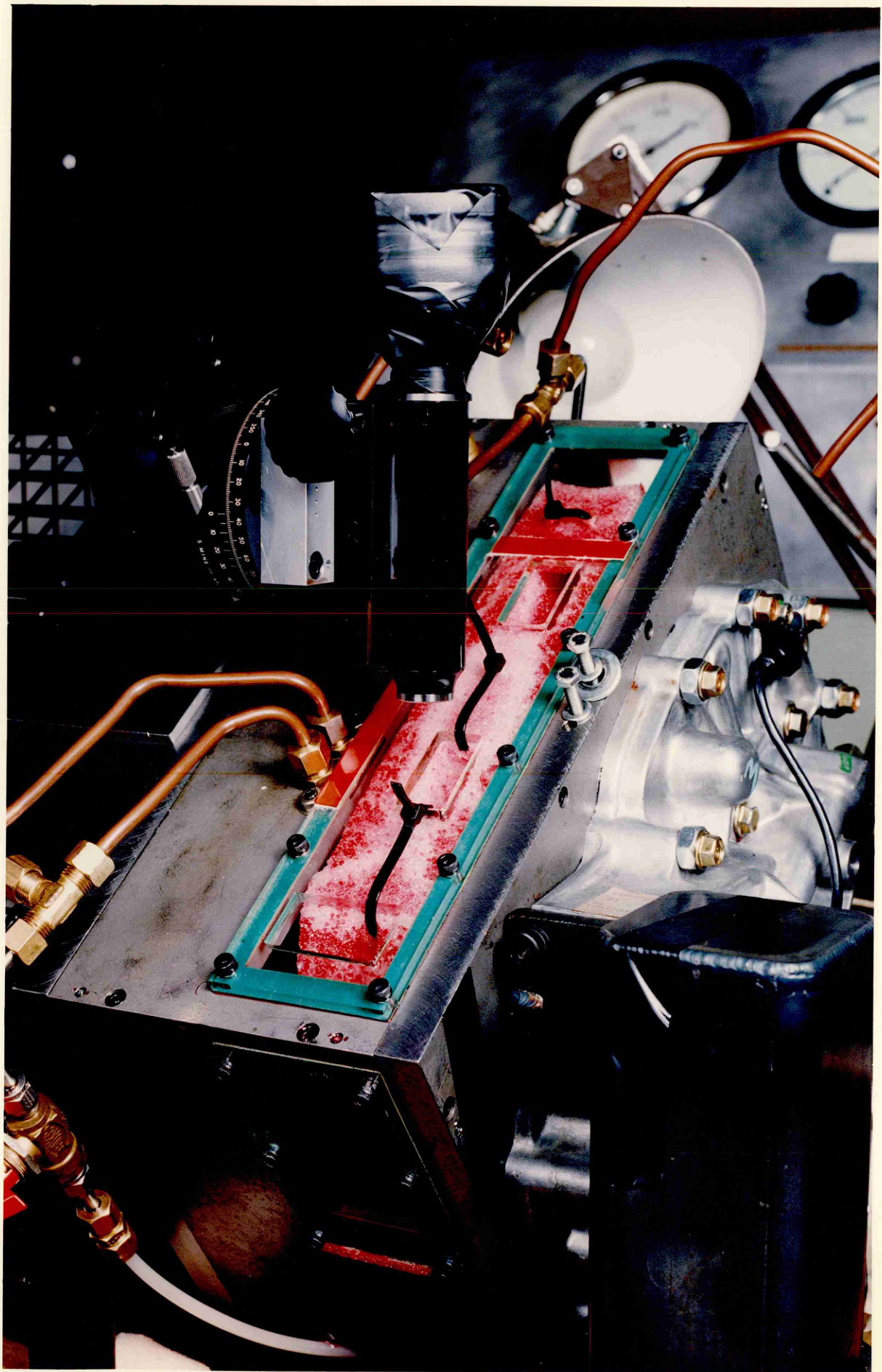
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OPTICAL MEASUREMENT OF BELT MISALIGNMENT

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CHAPTER 8

RESULTS

8.1 Introduction

This chapter shows the graphical results obtained from this work. The results have been divided into three main sections. The first section deals with the belt misalignment geometrical behaviour and values; the second deals with belt slip; while the third one deals with the efficiency of the transmission with regards to belt misalignment. The calibration curves from where some of these results were derived are shown in the Appendixes.

8.2 Belt Misalignment (Fig. 8.1 to 8.11)

These figures are shown in sequence in terms of their derivation. That is, the information from the preceding figure/s is/are required to produce the next except for the last two figures, Fig. 8.10 and 8.11 were put in for convenience to show at what primary pulley radii the two ideal straight planes and zero belt misalignment took place. The derivations started with the primary and secondary pulley radii relationship, Fig. 8.1 and was then followed by the belt position relative to the primary and secondary pulleys with respect to the transmission speed ratio, Fig. 8.2. This figure represents the values for $\delta = 0.00 \text{ mm}$. That is, the pulleys original set up positions were maintained. From these two figures, the belt misalignment values were obtained, Fig. 8.3; while Fig. 8.4 and 8.5 depicted the plan view of the belt geometrical position along the compression straight, between the primary pulley exit from and the secondary pulley entry into their grooves. The rest of the figures, Fig. 8.6, 8.7, 8.8 and 8.9 are also shown in the same sequence as the previous one, except that this time they represent the results for $\delta = 2.00 \text{ mm}$. That is, the secondary fixed pulley half had been shifted by 2.00 mm along its shaft.

8.3 Belt Slip (Fig. 8.12 to 8.15)

These figures show the possibility of belt slipping taking place. They are shown in the form of the transmission speed ratios. That is, the one measured using the speed ratios

of the transmission pulleys radii $\frac{RB_s}{RB_p}$, as compared to the one measured using the transmission pulleys speed ratio $\frac{\omega_p}{\omega_s}$.

If there is no belt slip then,

$$\frac{\omega_p}{\omega_s} = \frac{RB_s}{RB_p}$$

that is, they should be equal, but if there was a belt slip then,

$$\frac{\omega_p}{\omega_s} > \frac{RB_s}{RB_p}$$

which is the speed ratio based on the pulley angular speeds which will be greater than the one based on pulleys radii. The pulleys radii in this work, were measured using the optical (opt.) instrumentation while the pulley angular speeds were measured using the torque and speed meter.

The first two figures, Fig. 8.12 and 8.13 represent the one with $\delta = 0.00 \text{ mm}$, while the last two Fig. 8.14 and 8.15 for $\delta = 2.00 \text{ mm}$. Fig. 8.12 and 8.14 were carried out at constant input torque, while Fig. 8.13 and 8.15 at constant output torque.

8.4 Transmission Efficiency (Fig. 8.16 to 8.31)

These figures show the results from all the 20 experiments carried out to study the transmission performance with regards to the pulley belt misalignment values. Please refer back to Table 7.1 for the brief summary on the types of experiment carried out. In short, Expt. 4a represents the experiments carried out with $\delta = 0.00 \text{ mm}$ and Expt. 4b represents the experiments carried out with $\delta = 2.00 \text{ mm}$; the odd number following the alphabet represents constant input torque case while the even number represents constant output torque. However, the results here are presented in a different format as that of Table 7.1 for convenience of the discussion later in Chapter 9. They are divided into two main categories that is, constant input torque (Fig. 8.16 to 8.24) and constant output torque (Fig. 8.25 to 8.31).

8.4.1 Constant Input Torque (Fig. 8.16 to 8.23)

Fig. 8.16 shows the comparison of the transmission efficiency as the input or primary pulley speeds were varied from about 625 to 1750 *rpm* for Expt. 4a11, 4b11, 4a12 and 4b12; while Fig. 8.17 shows their calculated results.

Fig. 8.18 shows the comparison of the transmission efficiency as the speed ratio varies from overdrive to underdrive position for input or primary pulley speed of about 825 *rpm* for Expt. 4a31 and 4b31; while Fig. 8.19 shows their calculated results.

Fig. 8.20 and 8.21 for Expt. 4a32 and 4b32 are similar to Fig. 8.18 and 8.19 respectively, but with an input or primary pulley speed of about 1250 *rpm*.

Fig. 8.22 and 8.23 for Expt. 4a32 and 4b32 are also similar to Fig. 8.18 and 8.19 respectively, but with an input or primary pulley speed of about 1750 *rpm*.

8.4.2 Constant Output Torque (Fig. 8.24 to 8.31)

Fig. 8.24 and 8.25 for Expt. 4a21, 4b21, 4a22 and 4b22 are similar to Fig. 8.16 and 8.17, respectively.

Fig. 8.26 and 8.27 for Expt. 4a41 and 4b41 are similar to Fig 8.18 and 8.19, respectively.

Fig. 8.28 and 8.29 for Expt. 4a42 and 4b42 are similar to Fig 8.20 and 8.21, respectively.

Fig. 8.30 and 8.31 for Expt. 4a43 and 4b43 are similar to Fig 8.22 and 8.23, respectively.

8.5 Figures

Fig. 8.1 Pulley radii graphs

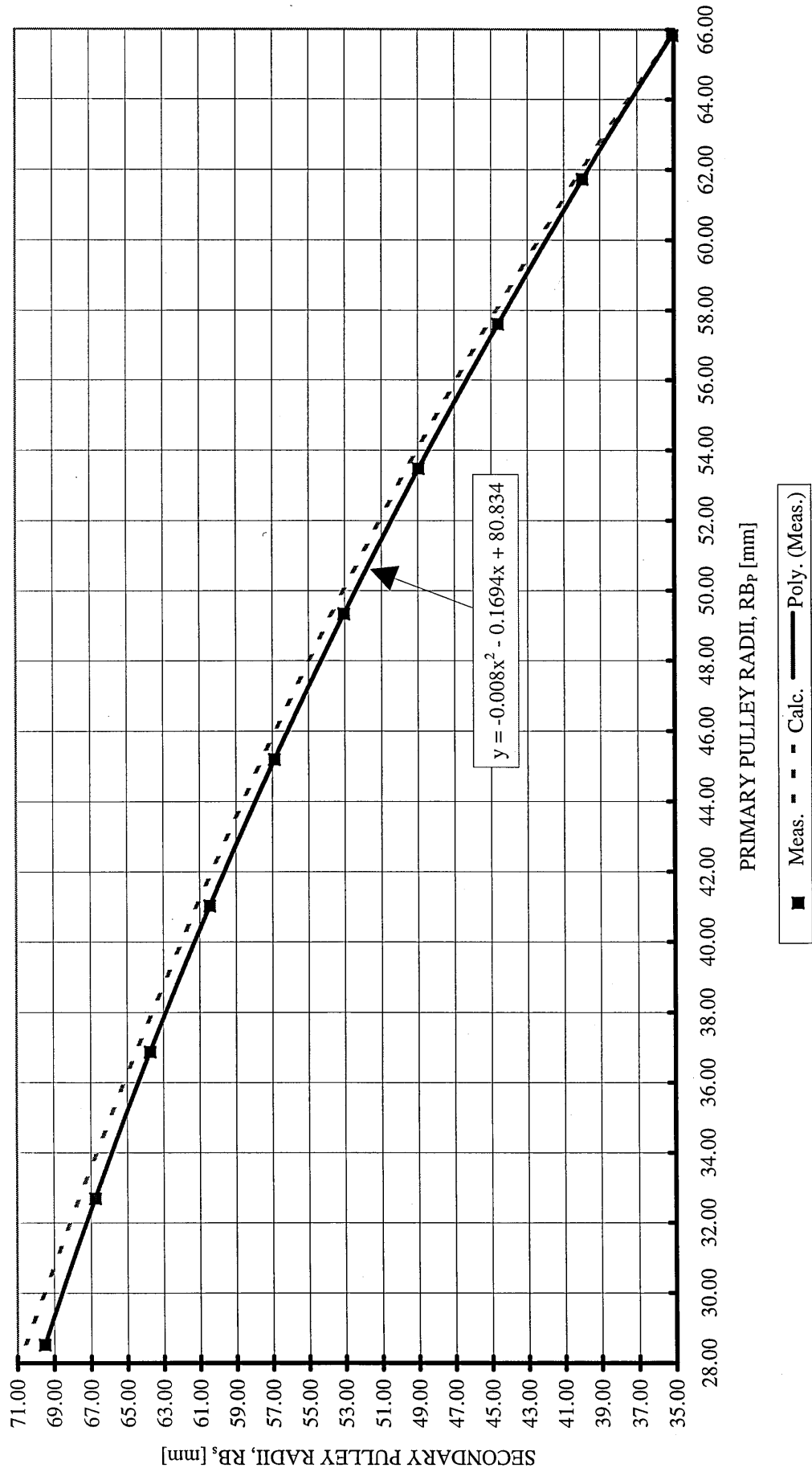


Fig. 8.2 Belt axial movement graphs against speed ratio (adelta = 0.00 mm)

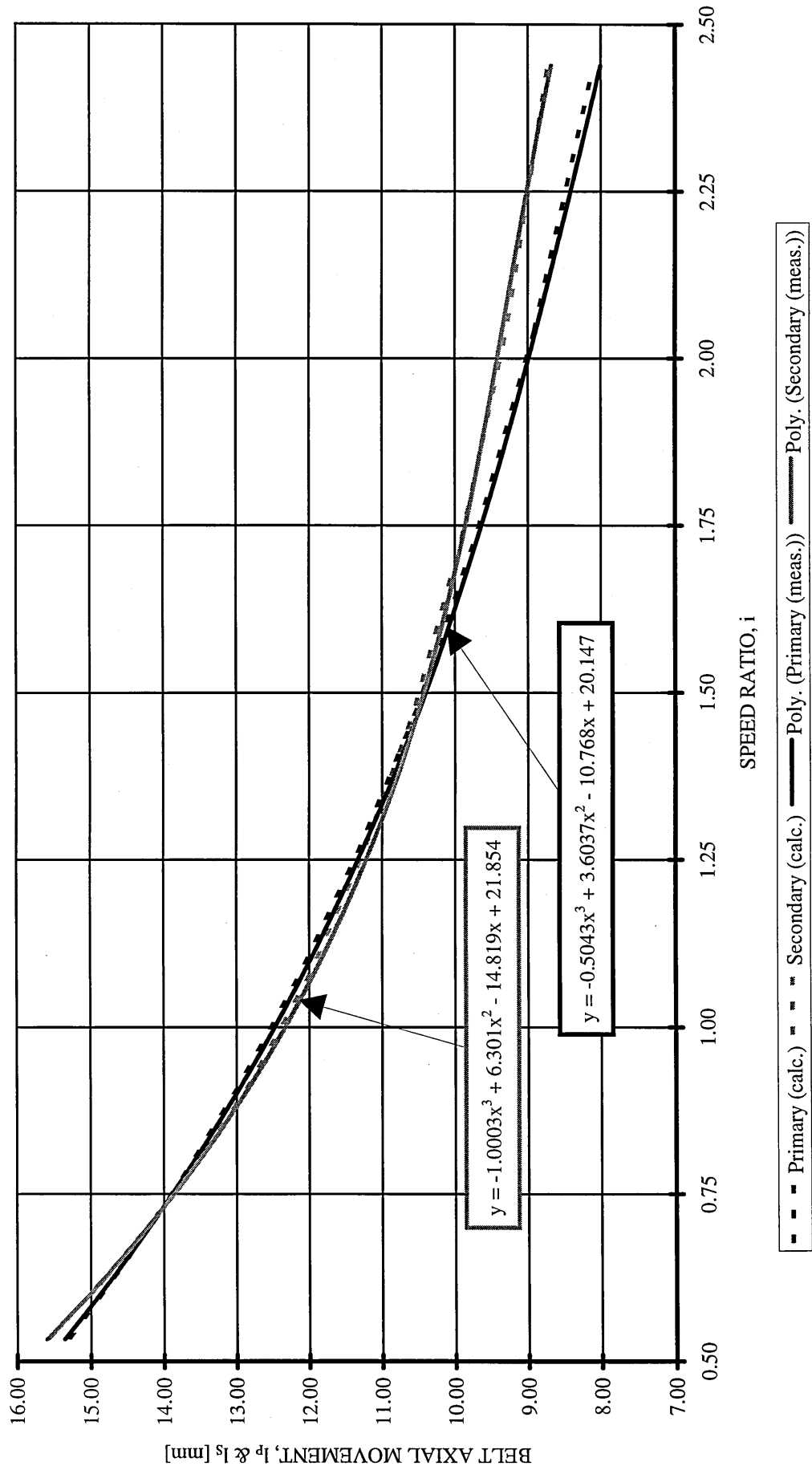


Fig. 8.3 Belt misalignment graphs against speed ratio ($\delta = 0.00$ mm)

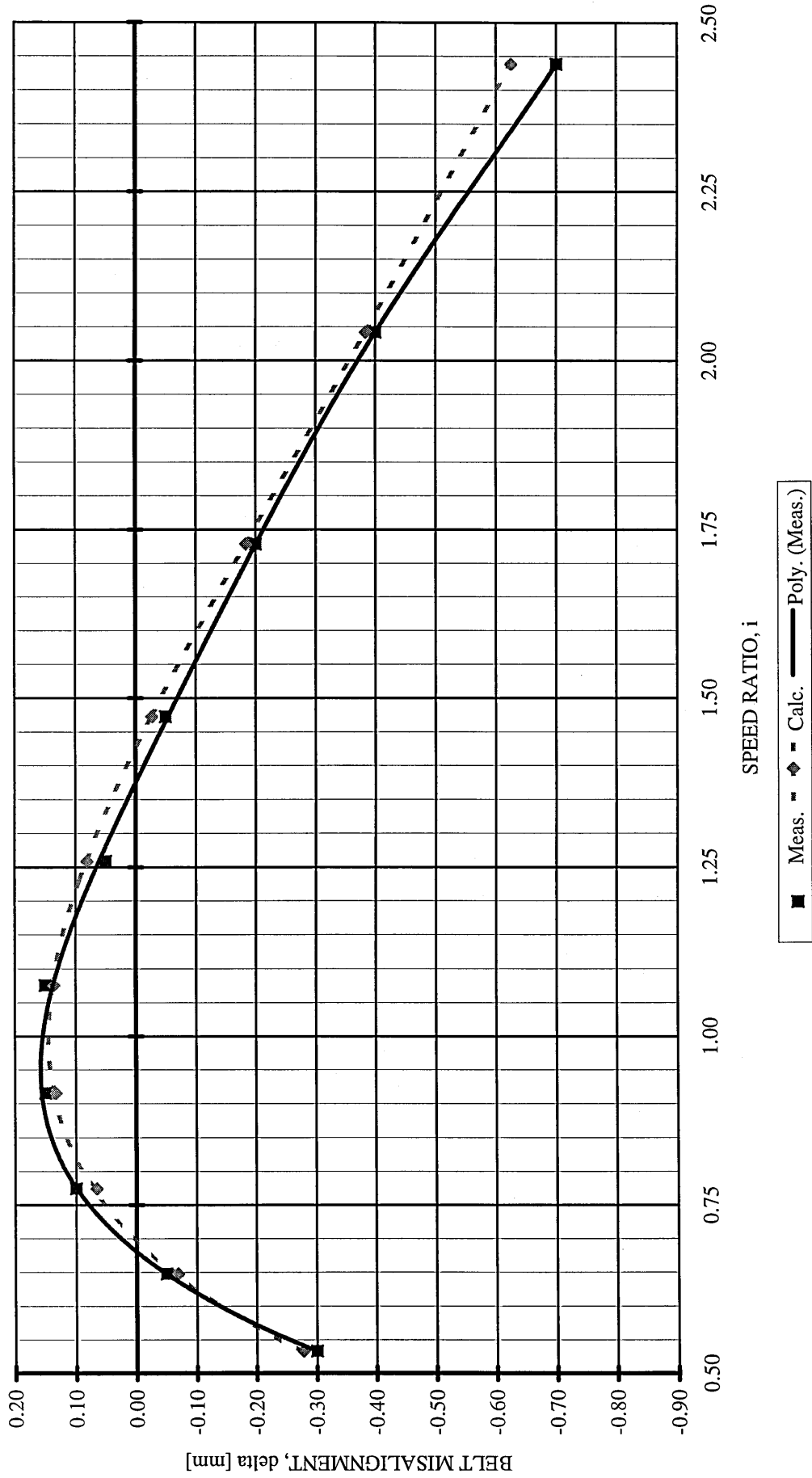


Fig. 8.4 Belt Misalignment Position (Measured, $\Delta = 0.00$ mm)

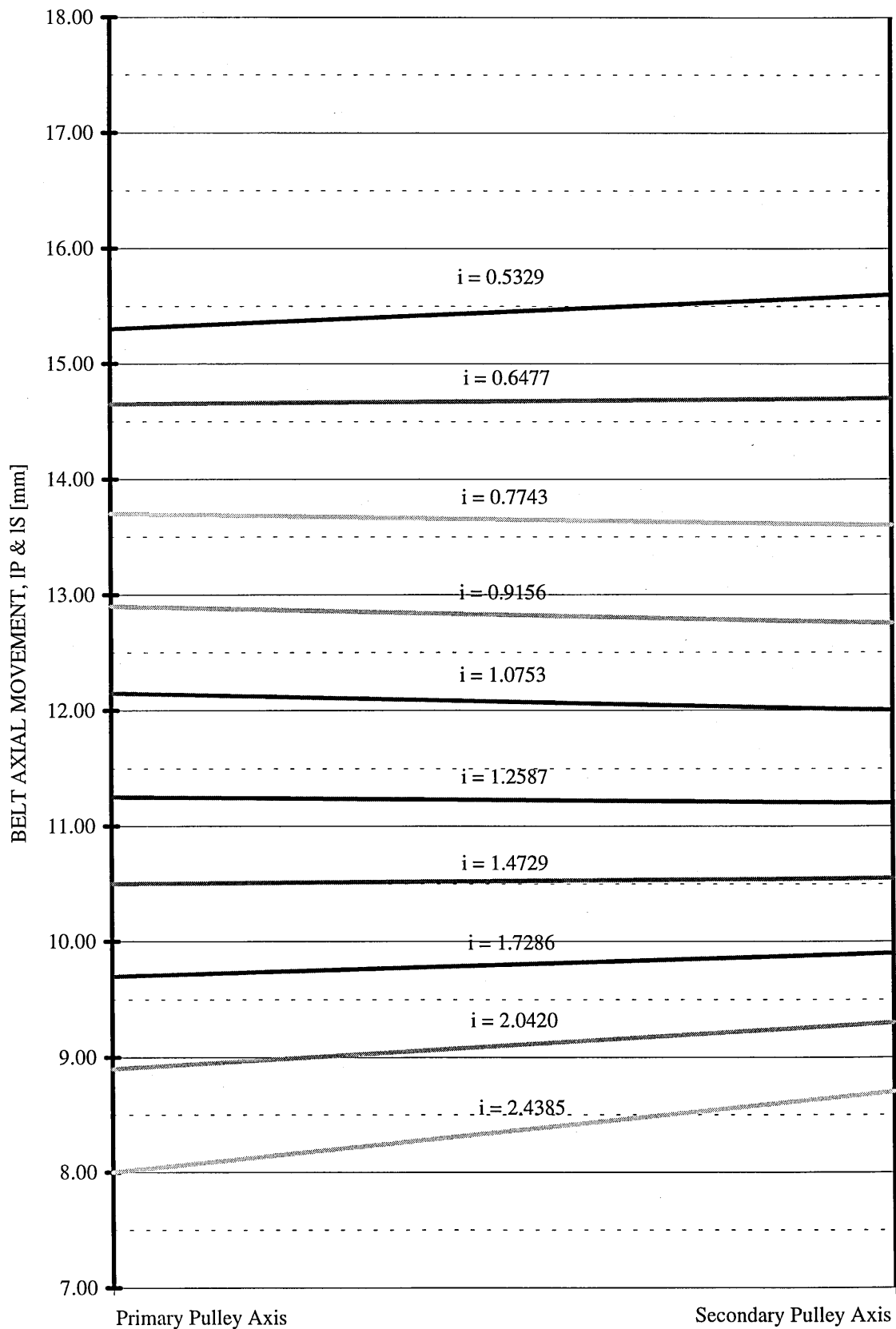


Fig. 8.5 Belt Misalignment Position (Calculated, $\Delta = 0.00$ mm)

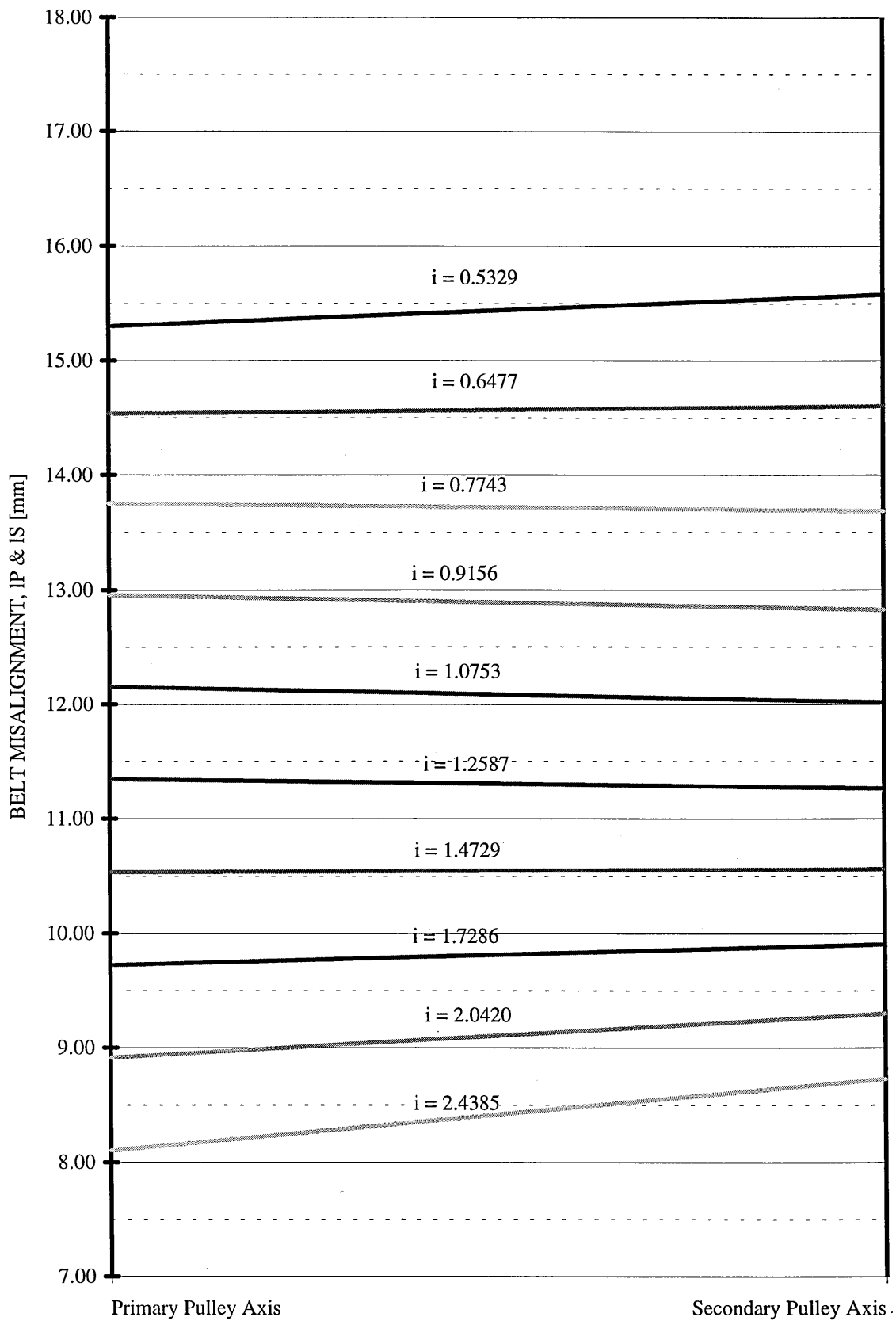


Fig. 8.6 Belt axial movement graphs against speed ratio (adelta = 2.00 mm)

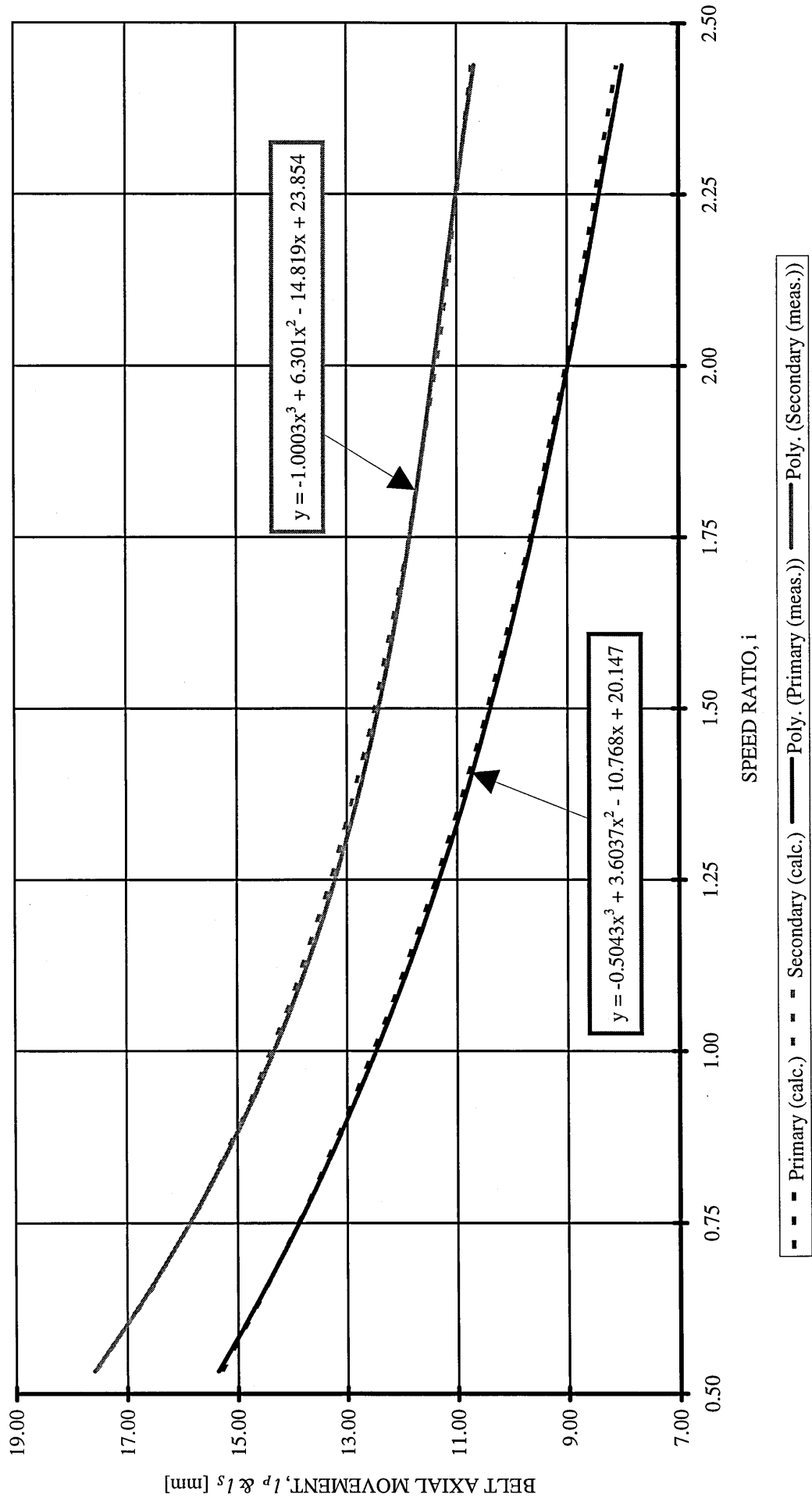


Fig. 8.7 Belt misalignment graphs against speed ratio ($\delta = 2.00$ mm)

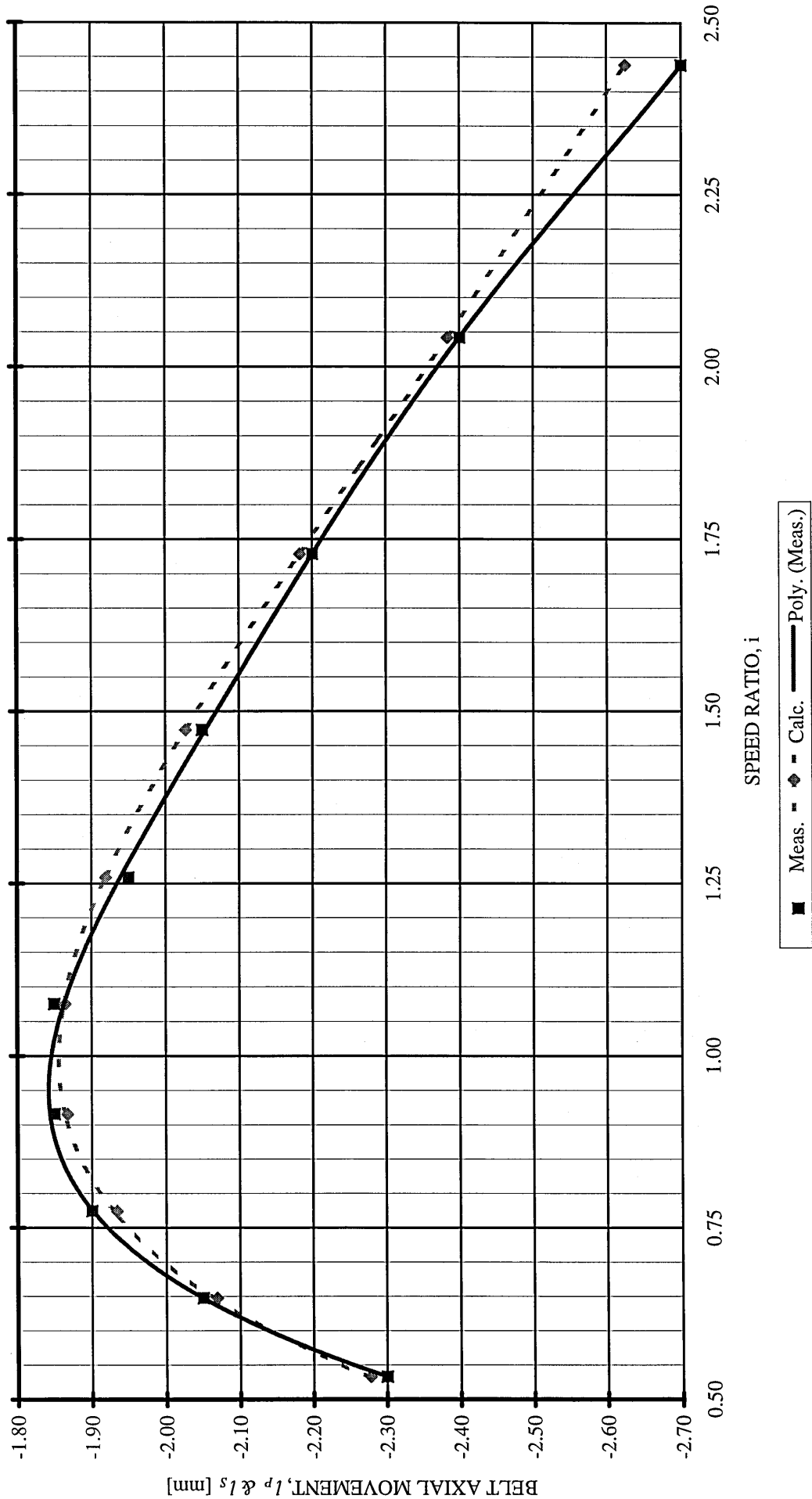


Fig. 8.8 Belt Misalignment Position (Measured, $\Delta = 2.00$ mm)

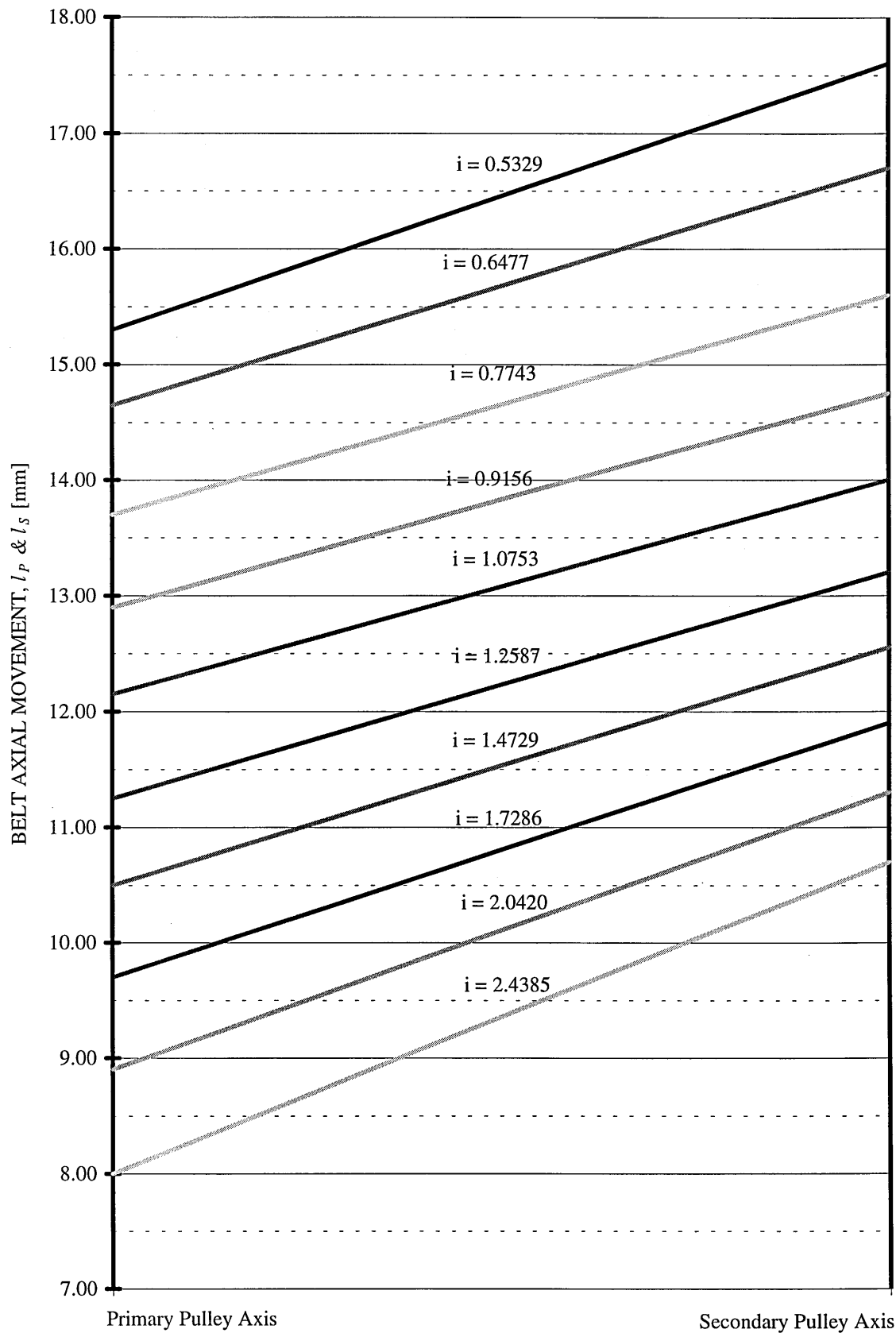


Fig. 8.9 Belt Misalignment Position (Calculated, $\delta = 2.00$ mm)

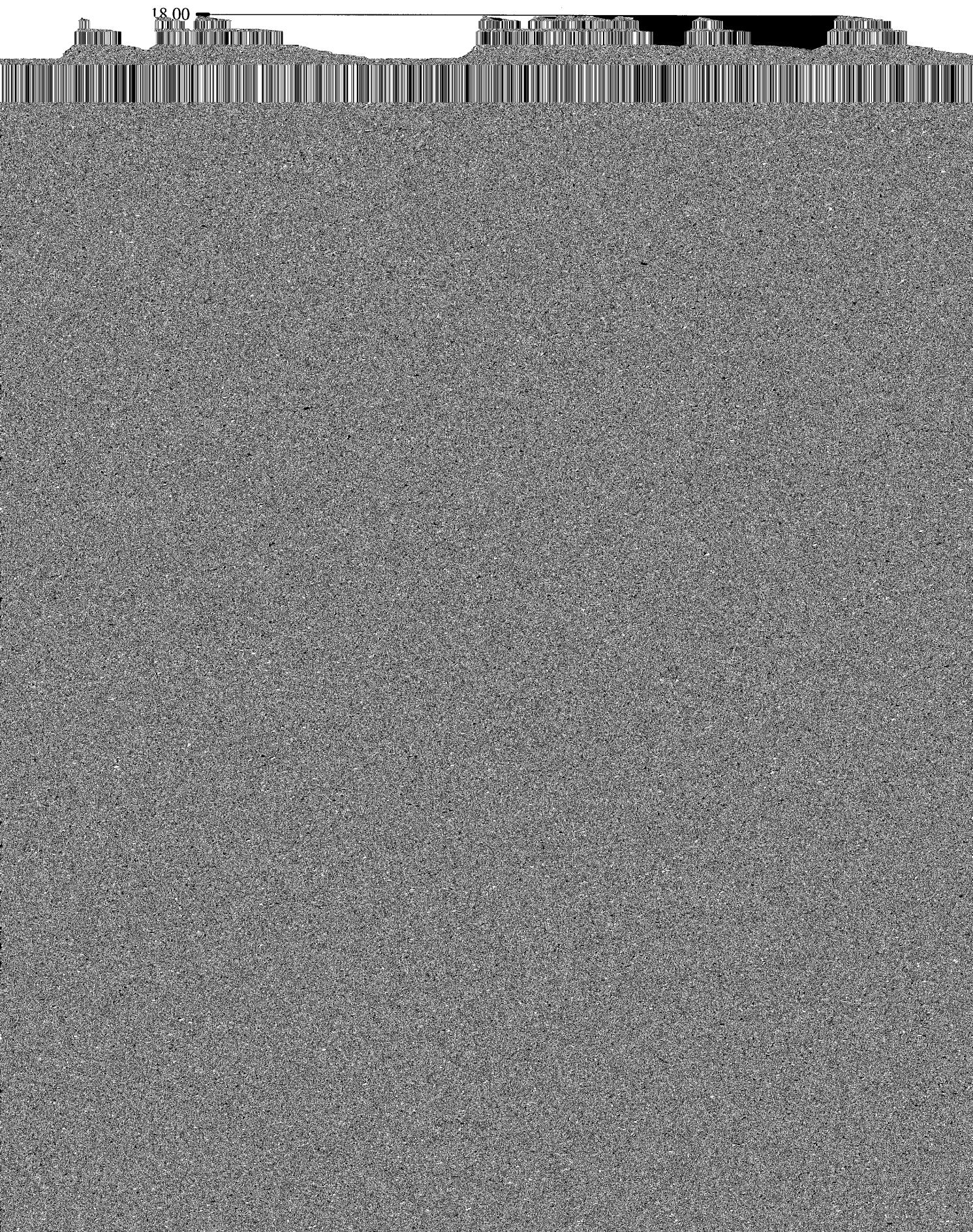


Fig. 8.10 Belt misalignment graphs against primary pulley radii ($\delta = 0.00$ mm)

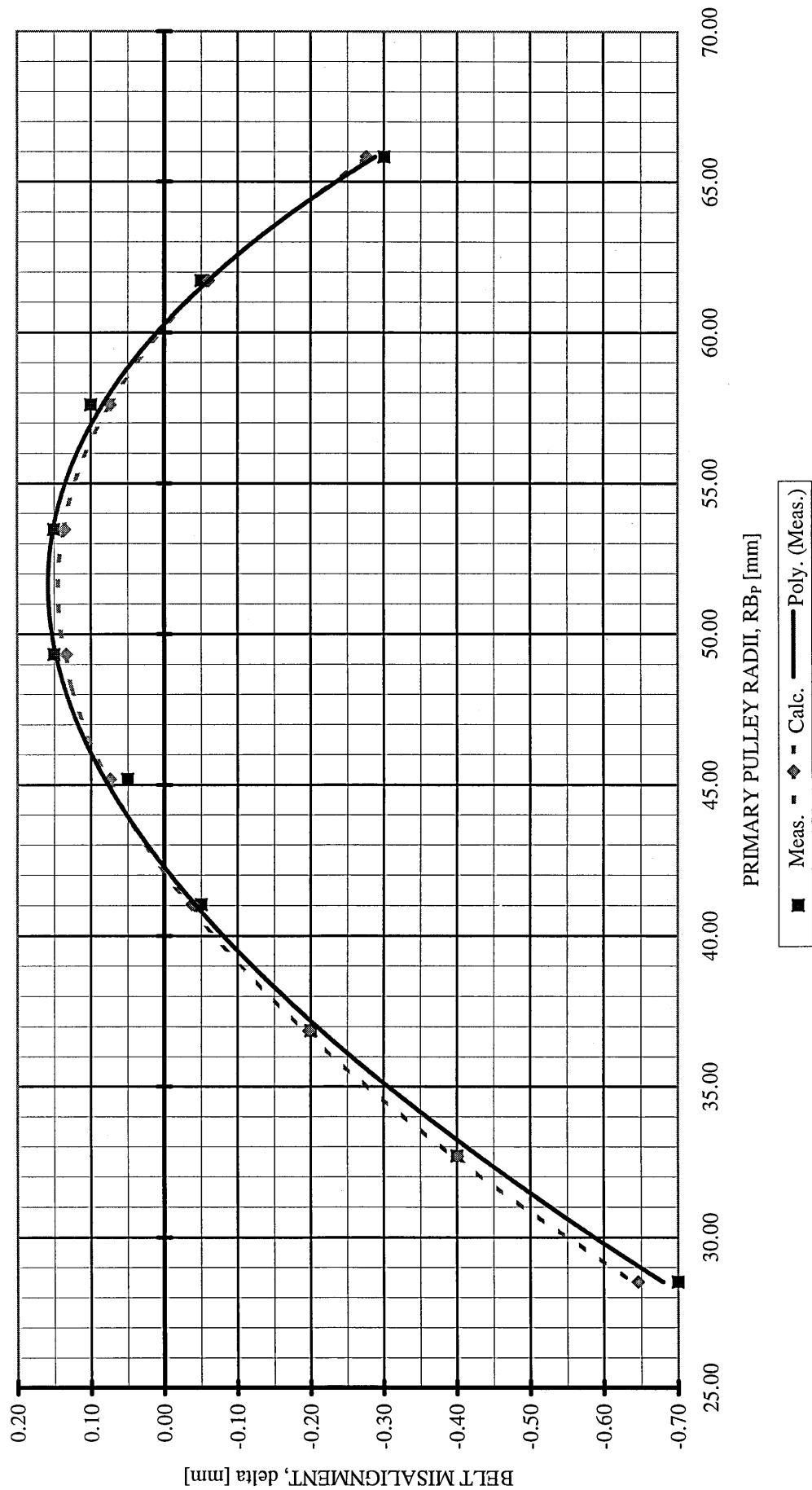


Fig. 8.11 Belt misalignment graphs against primary pulley radii (delta = 2.00 mm)

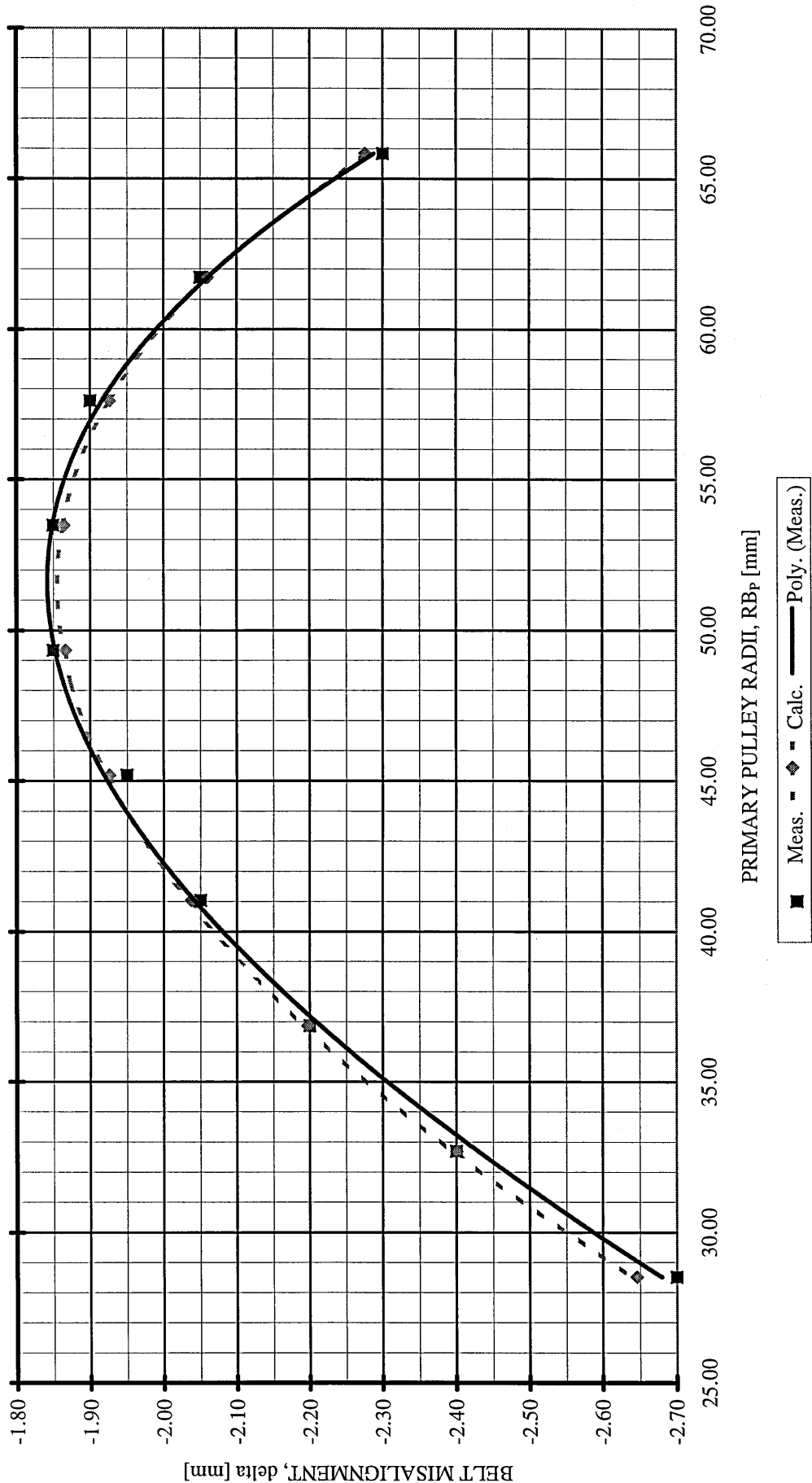


Fig. 8.12 Speed ratio curves (Constant input torque = 17 N.m, $\delta = 0.00\text{ mm}$ for Expt. 4a31, 4a32 and 4a33)

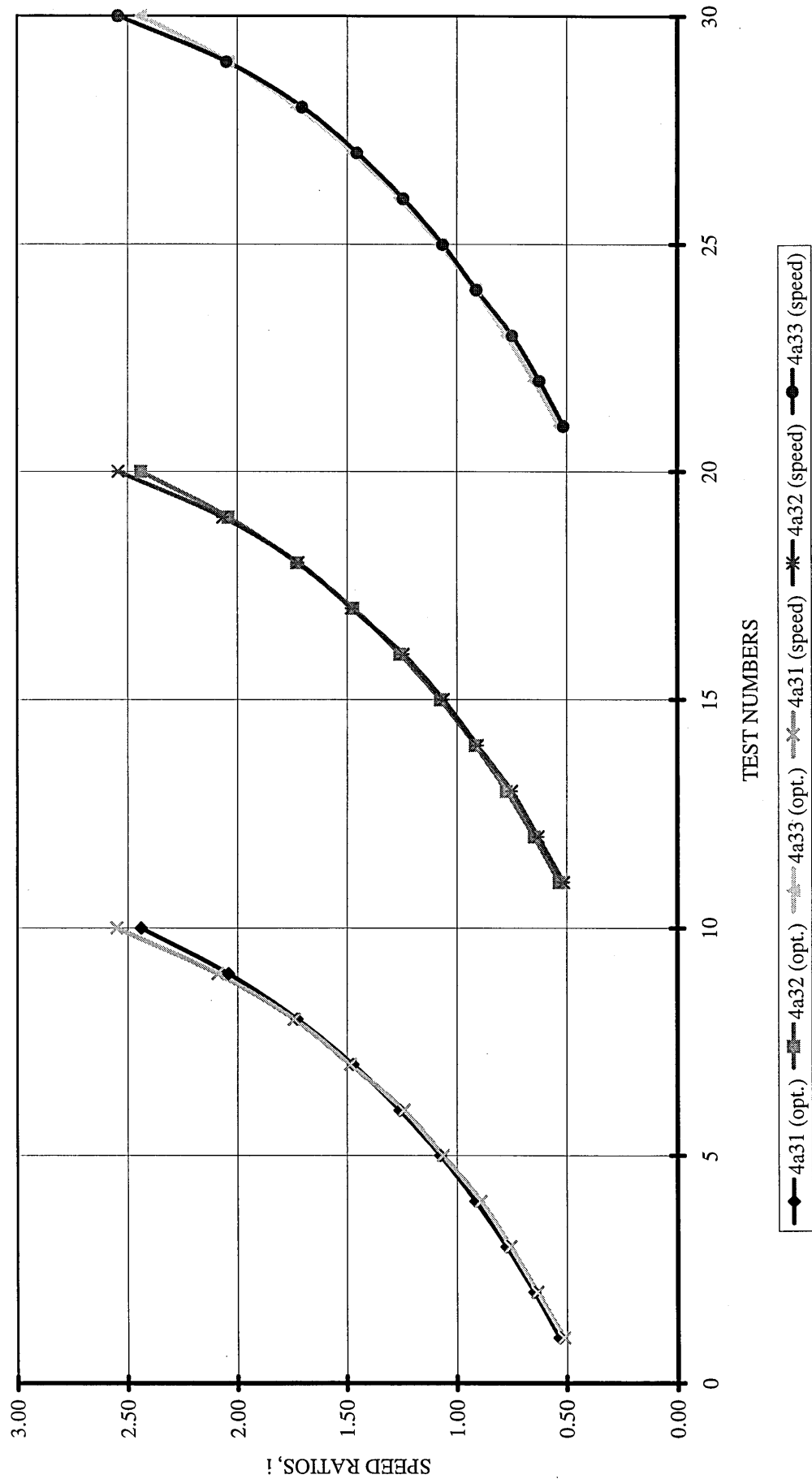


Fig. 8.13 Speed ratio curves (Constant input torque = 17 N.m, $\Delta = 2.00$ mm for Expt. 4b31, 4b32 and 4b33)

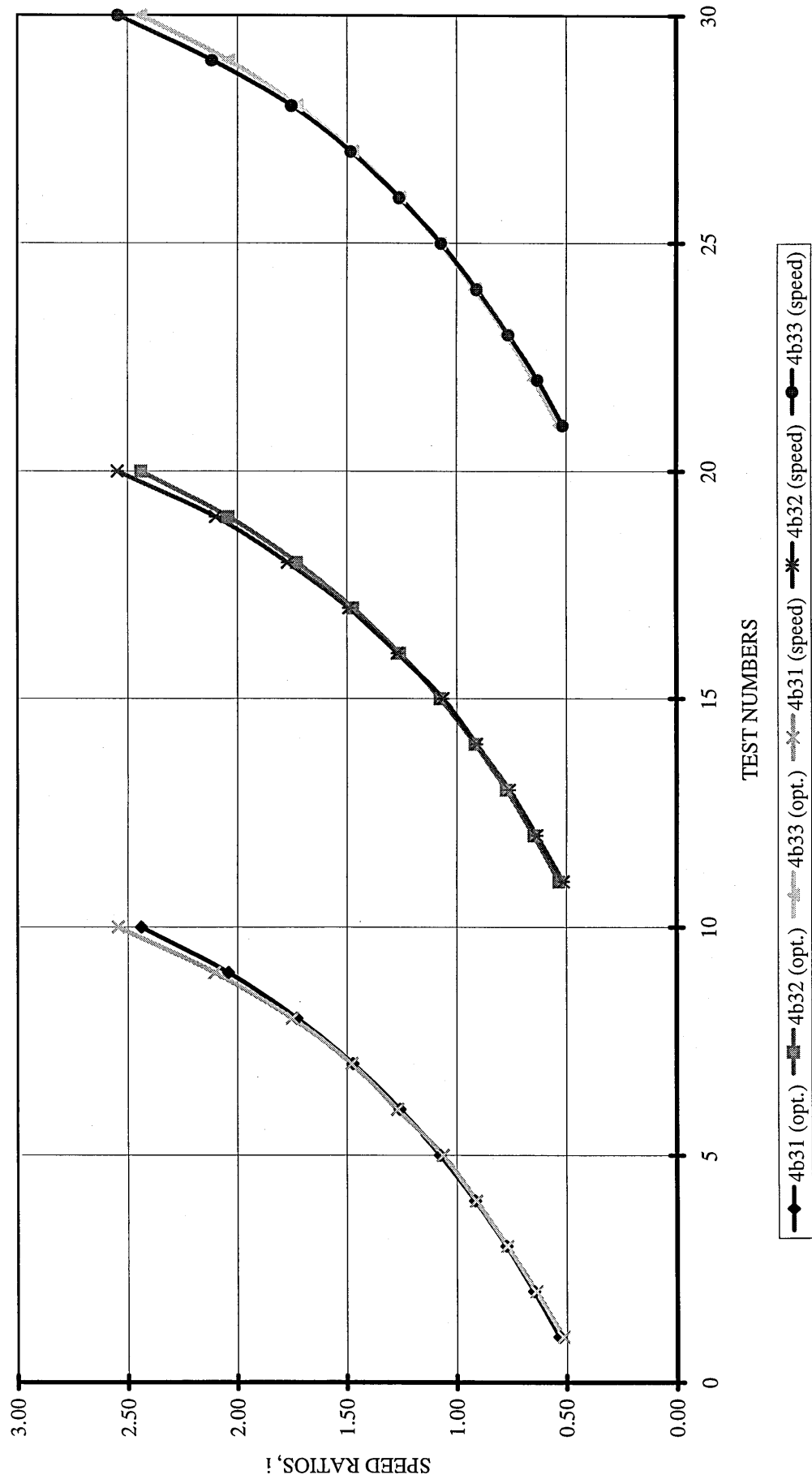


Fig. 8.14 Speed ratio curves (Constant output torque = 15 N.m, $\delta = 0.00\text{ mm}$ for Expt. 4a41, 4a32 and 4a43)

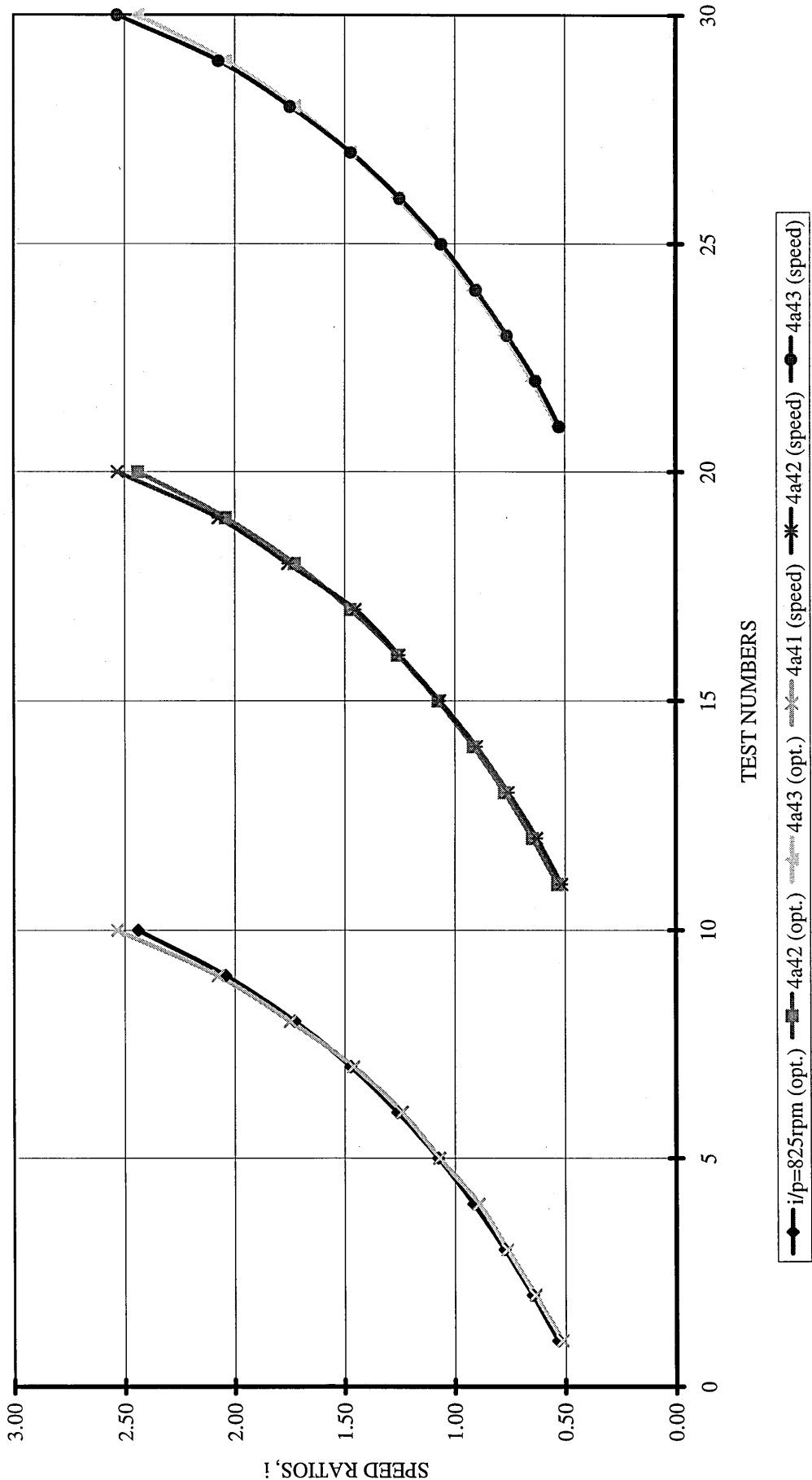


Fig. 8.15 Speed ratio curves (Constant output torque = 15 N.m, $\delta = 2.00\text{ mm}$ for Expt. 4b41, 4b42 and 4b43)

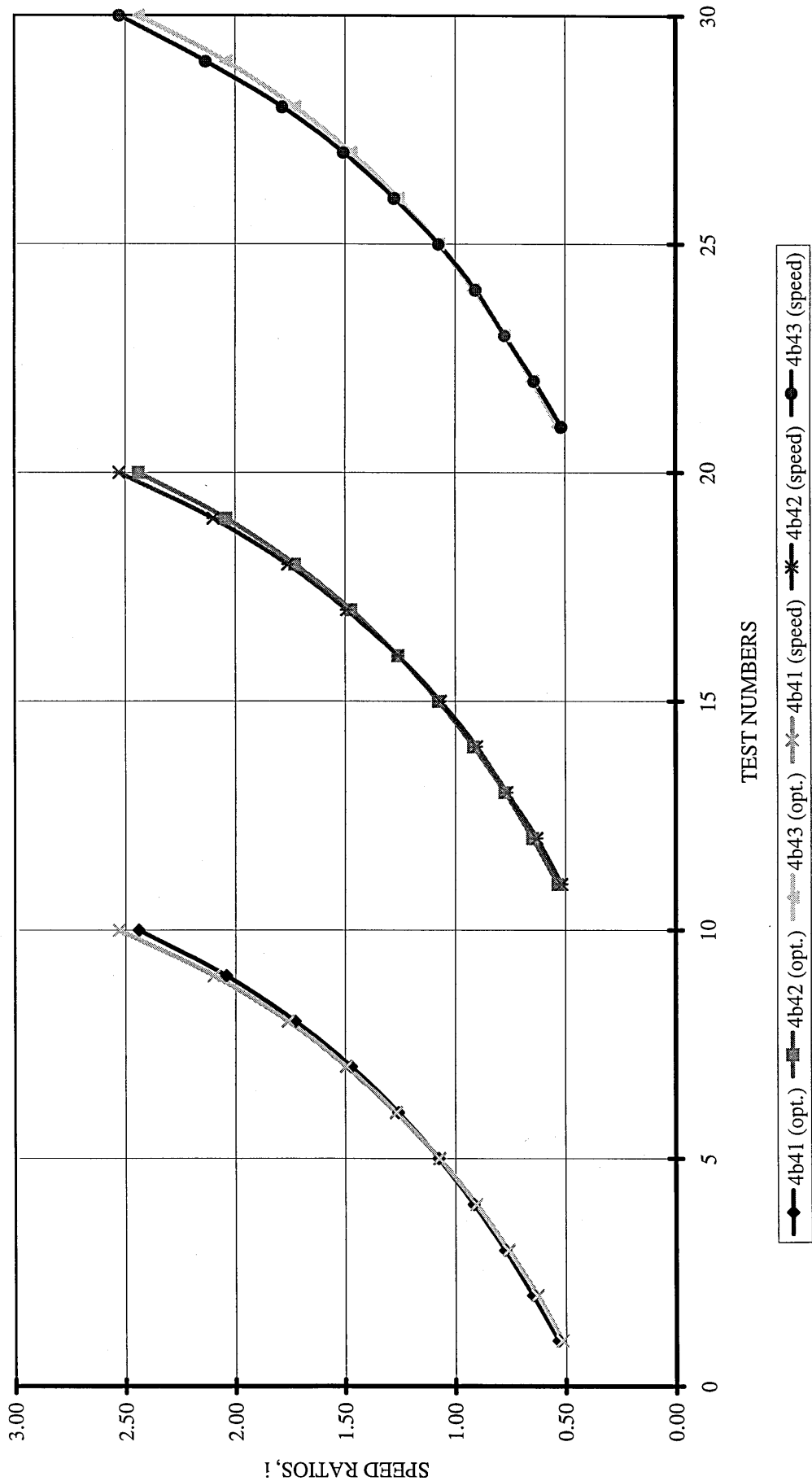


Fig. 8.16 Transmission efficiency (Measured, Constant input torque = 17 N.m)
- Expt. 4a11, 4a12, 4b11 and 4b12

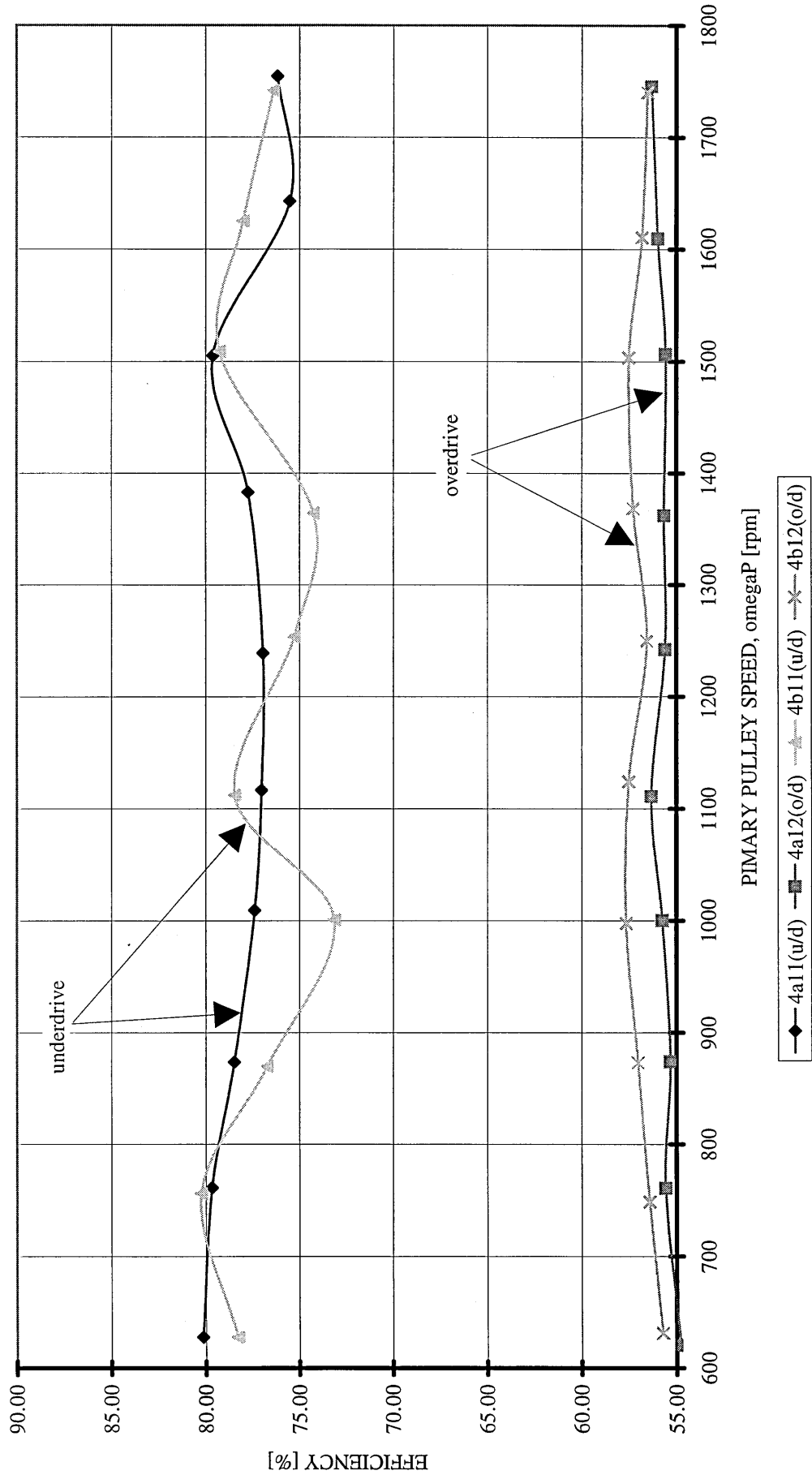


Fig. 8.17 Transmission efficiency (Calculated, Constant input torque = 17 N.m)
- Expt. 4a11, 4a12, 4b11 and 4b12

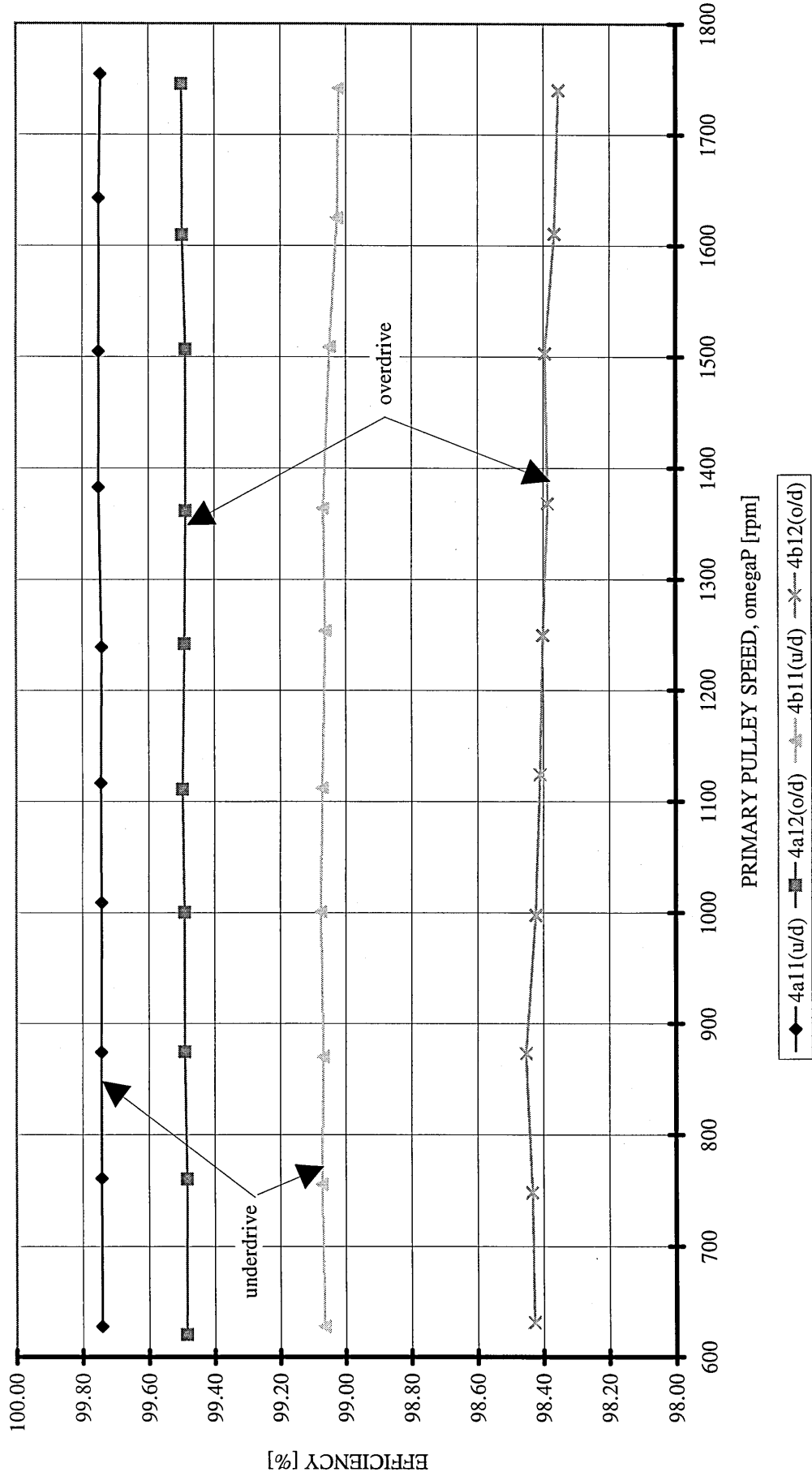


Figure 8.18 Transmission efficiency (Measured, Constant input torque = 17 N.m) - Expt. 4a31 and 4b31

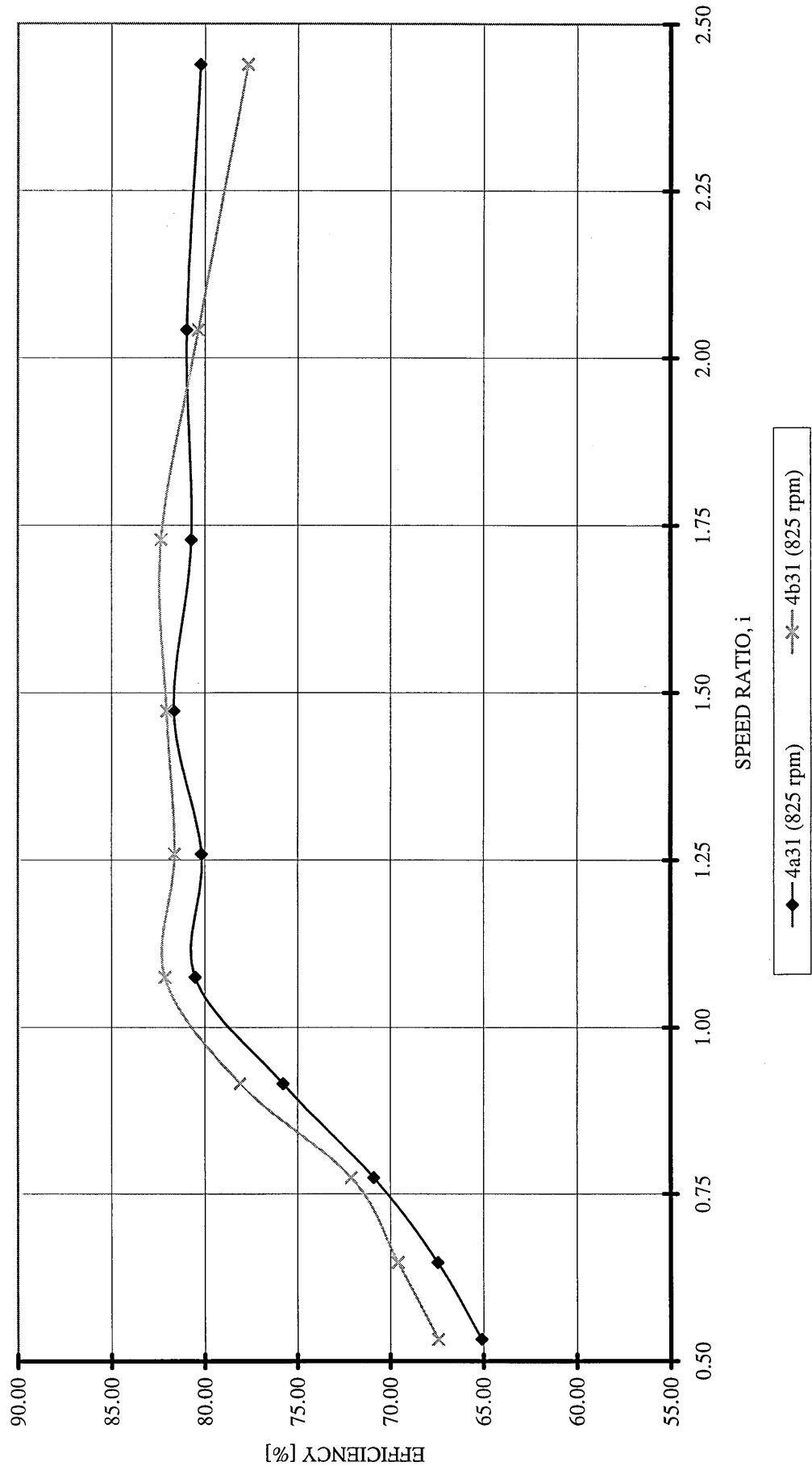


Figure 8.19 Transmission efficiency (Calculated, Constant input torque = 17 N.m) - Expt. 4a31 and 4b31

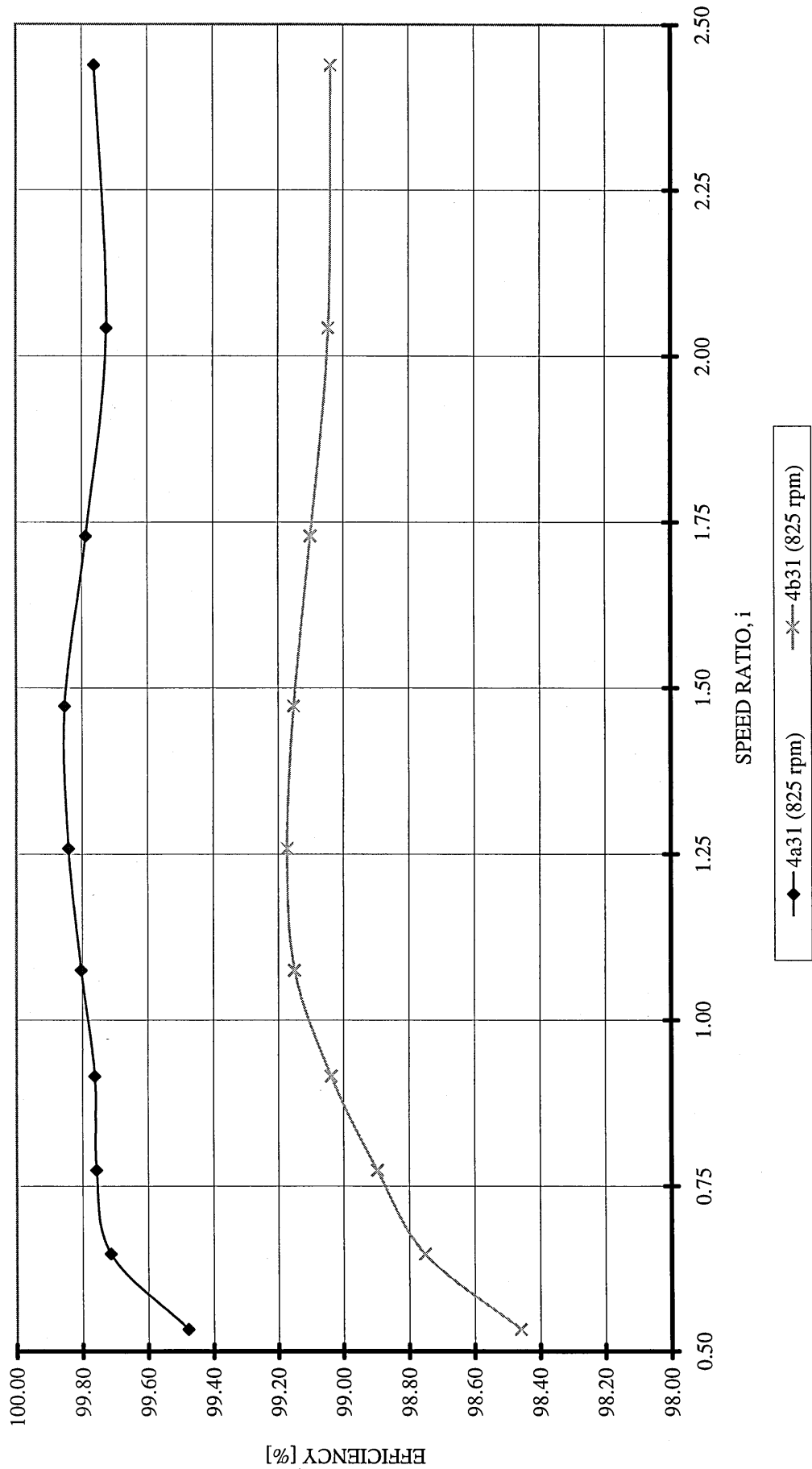


Figure 8.20 Transmission efficiency (Measured, Constant input torque = 17 N.m) - Expt. 4a32 and 4b32

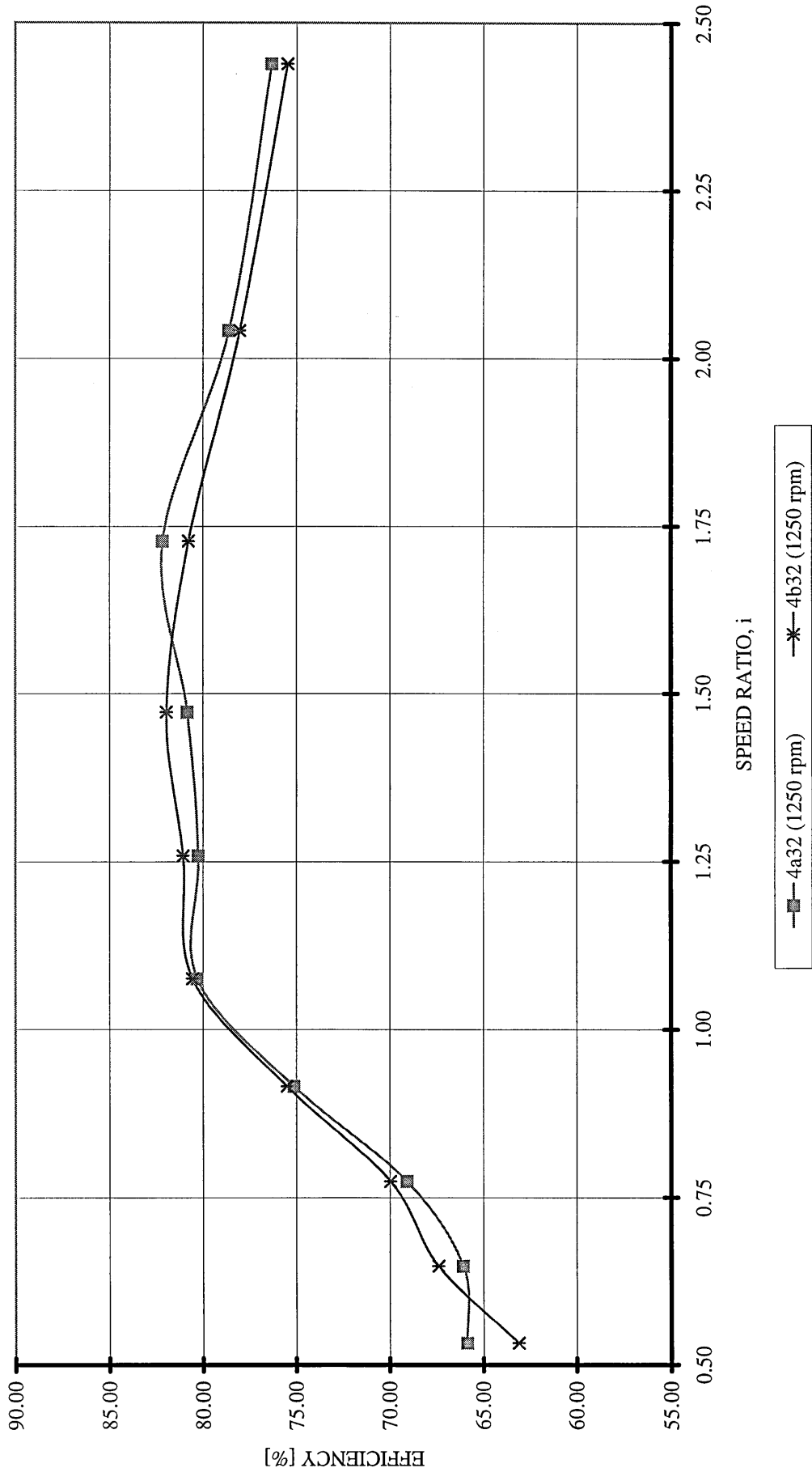


Figure 8.21 Transmission efficiency (Calculated, Constant input torque = 17 N.m) - Expt. 4a32 and 4b32

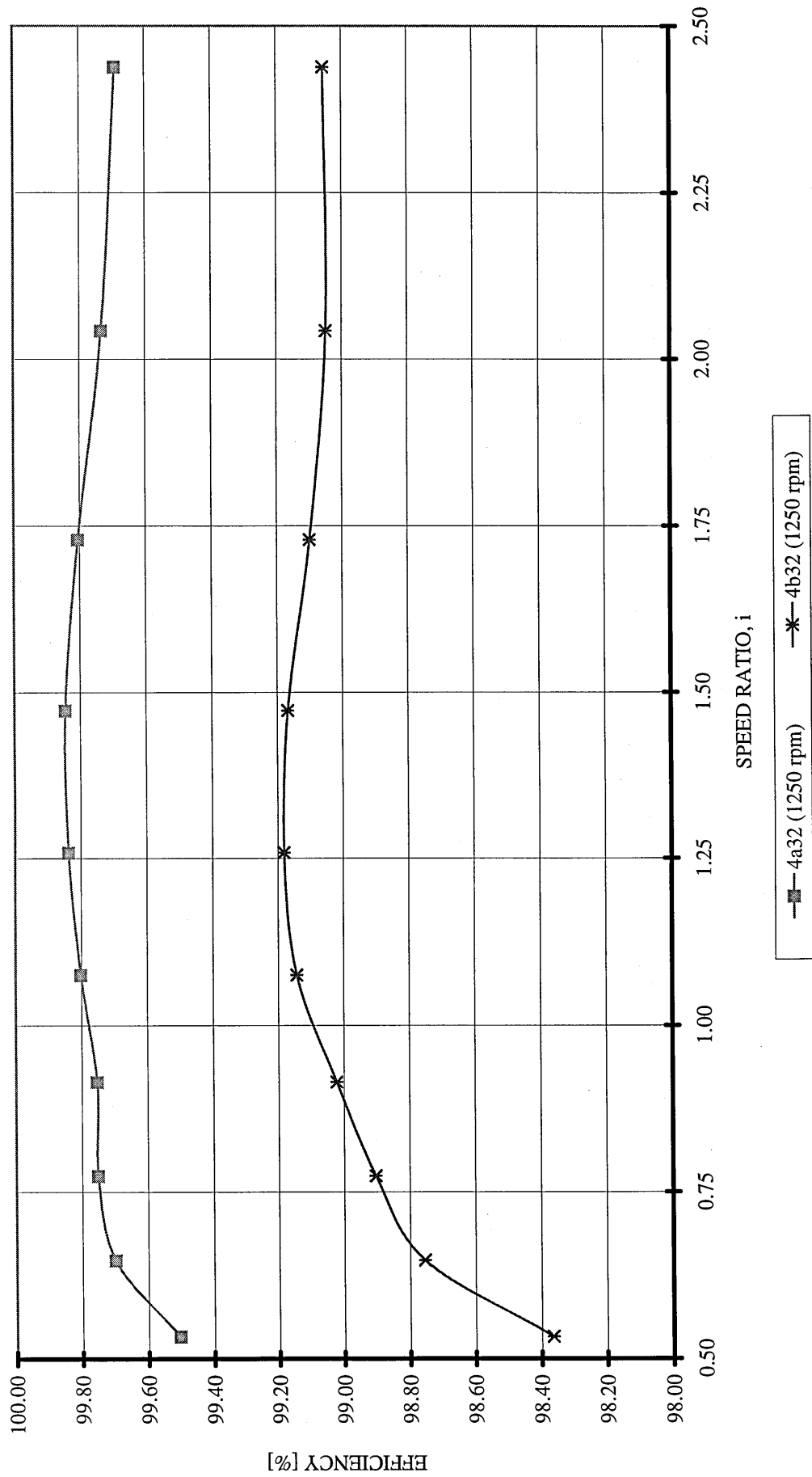


Figure 8.22 Transmission efficiency (Measured, Constant input torque = 17 N.m) - Expt. 4a33 and 4b33

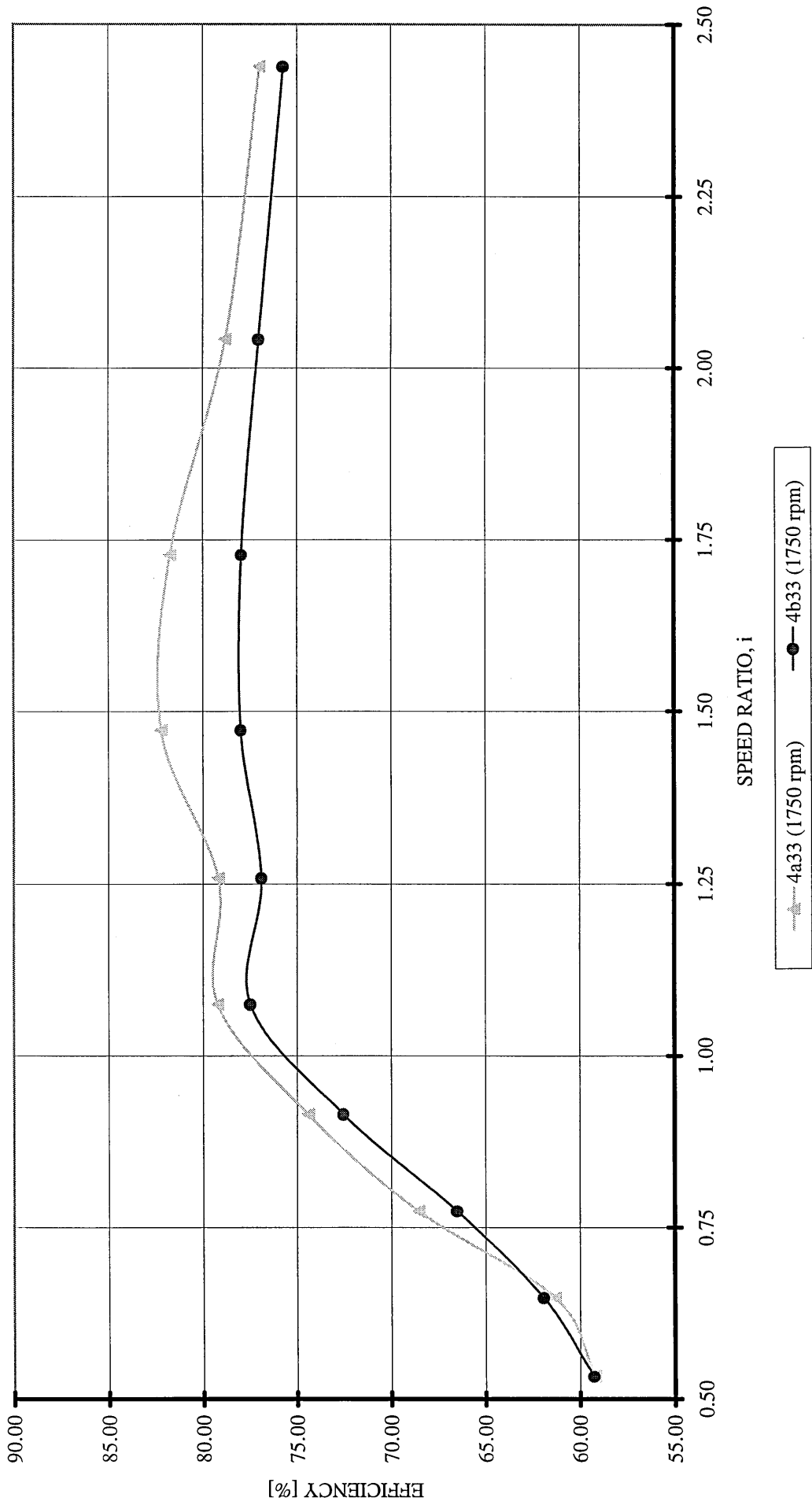


Figure 8.23 Transmission efficiency (Calculated, Constant input torque = 17 N.m) - Expt. 4a33 and 4b33

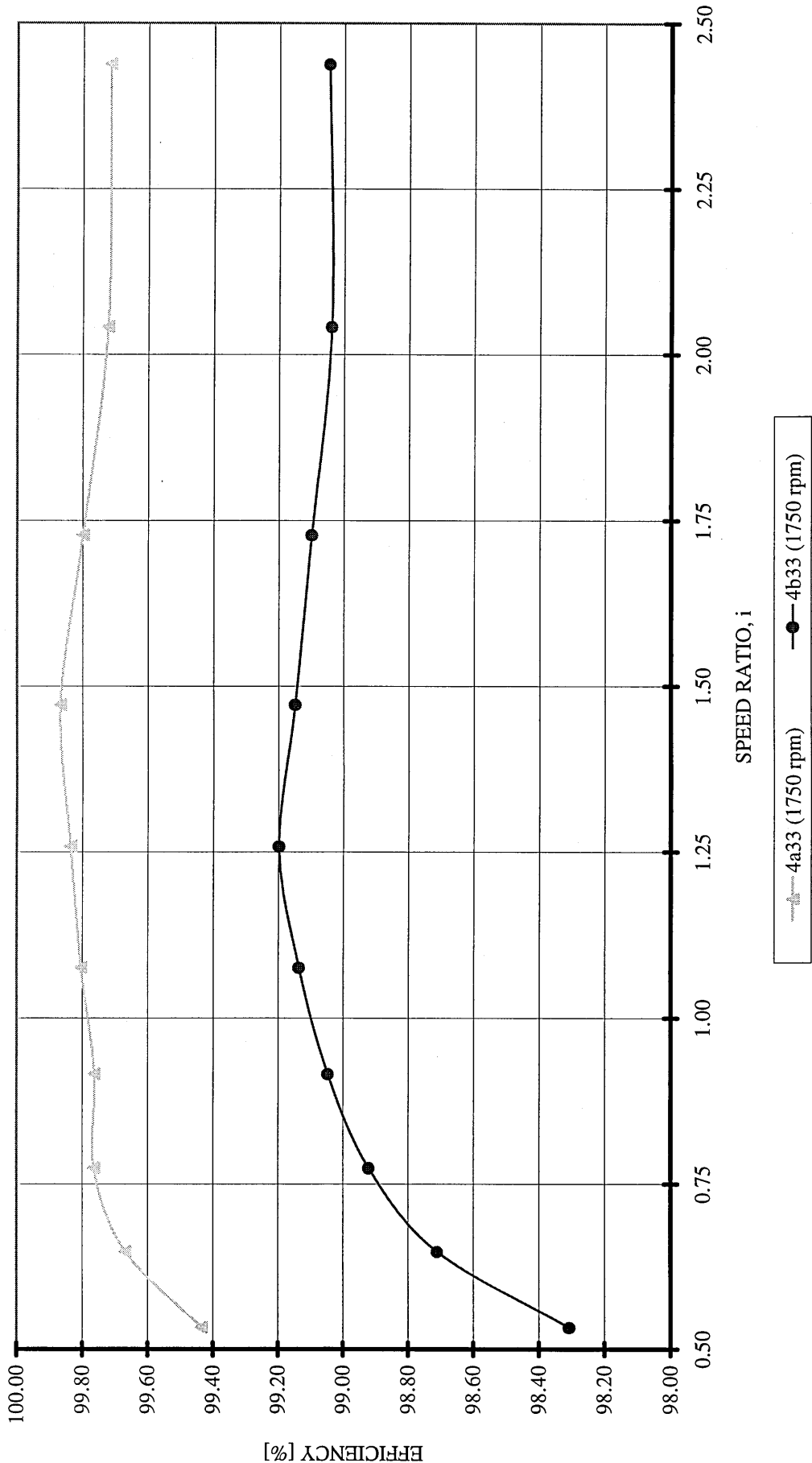


Fig. 8.24 Transmission efficiency (Measured, Constant output torque = 15 N.m)
- Expt. 4a21, 4a22, 4b21 and 4b22

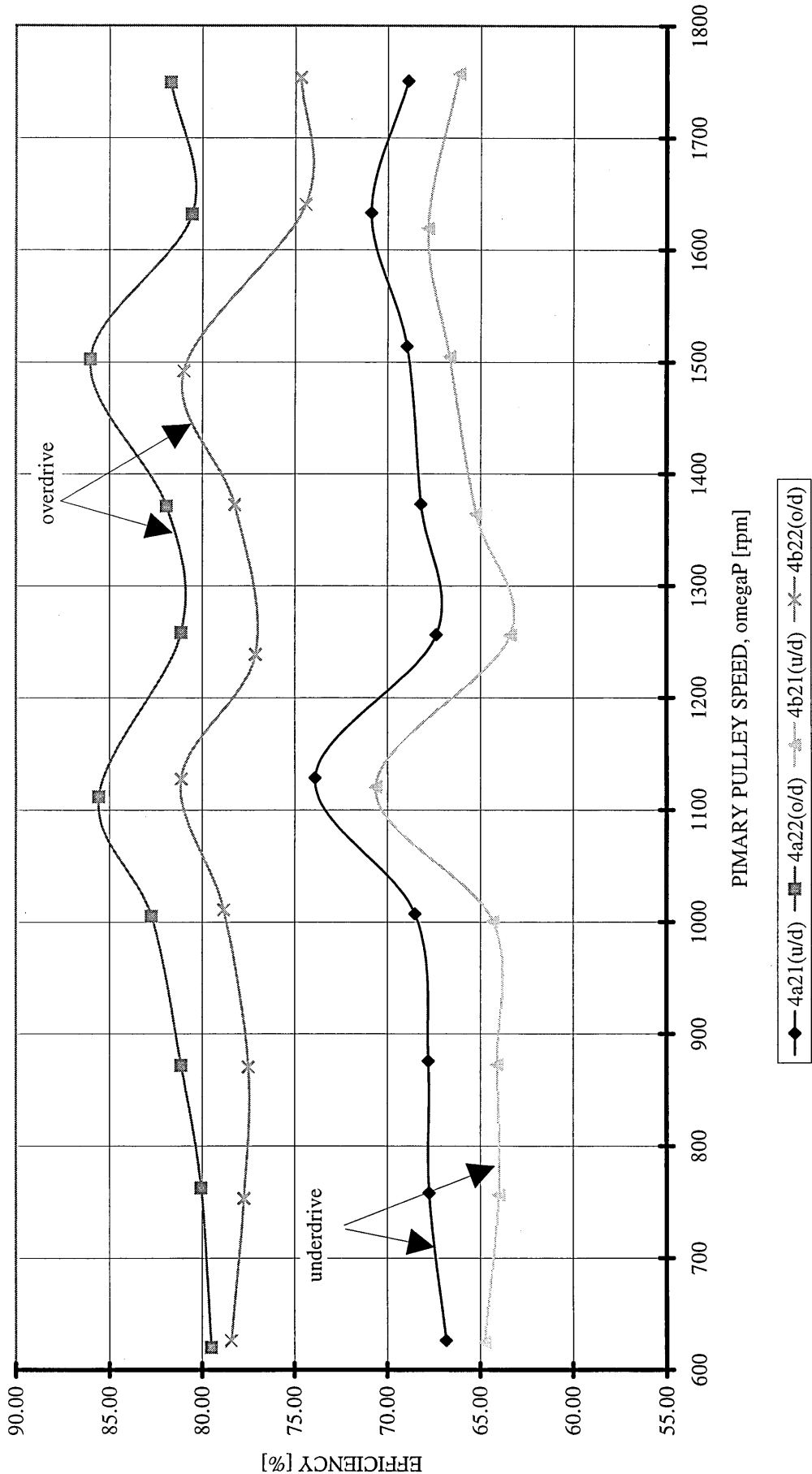


Fig. 8.25 Transmission efficiency (Calculated, Constant output torque = 15 N.m)
- Expt. 4a21, 4a22, 4b21 and 4b22

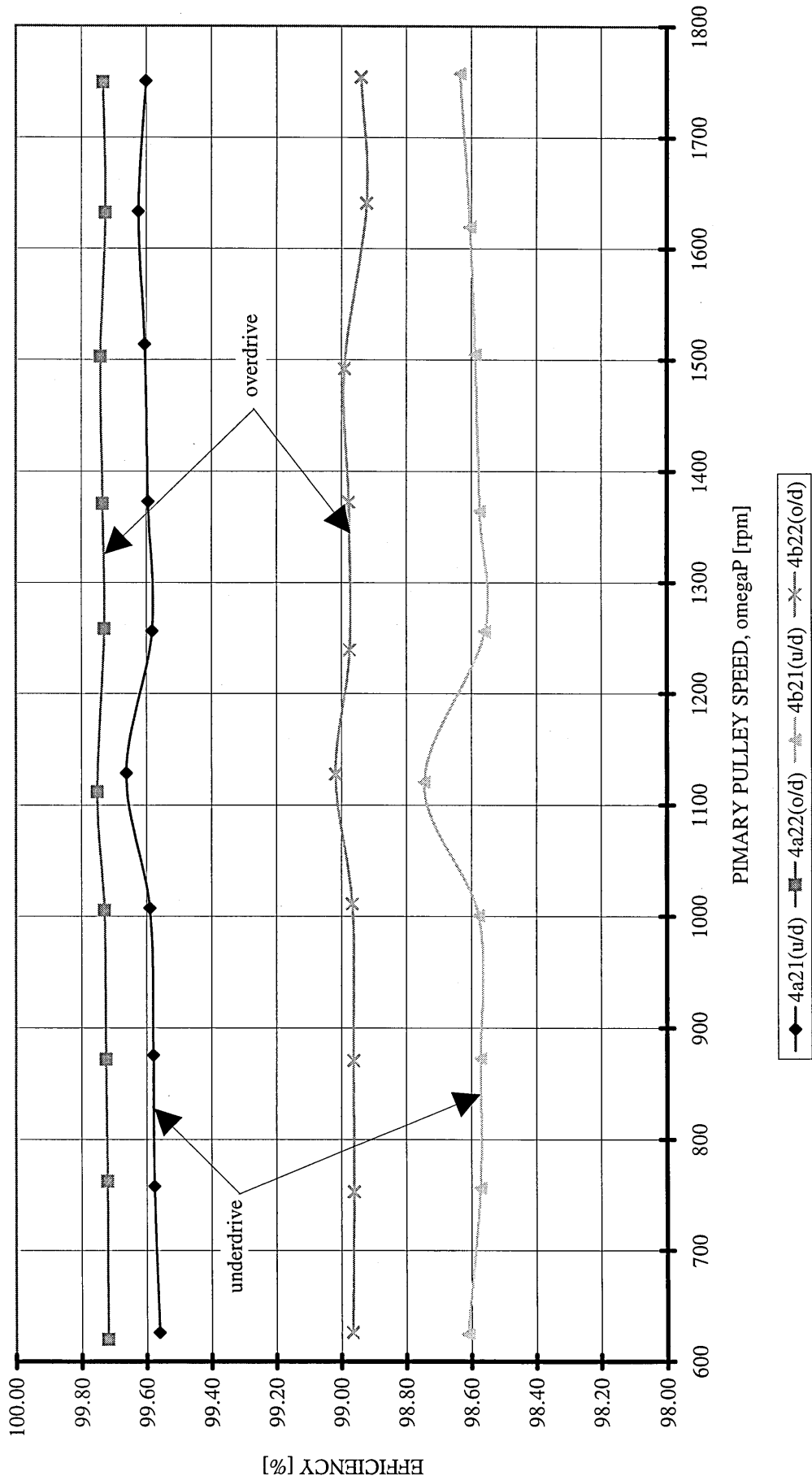


Figure 8.26 Transmission efficiency (Measured, Constant output torque = 15 N.m) - Expt. 4a41 and 4b41

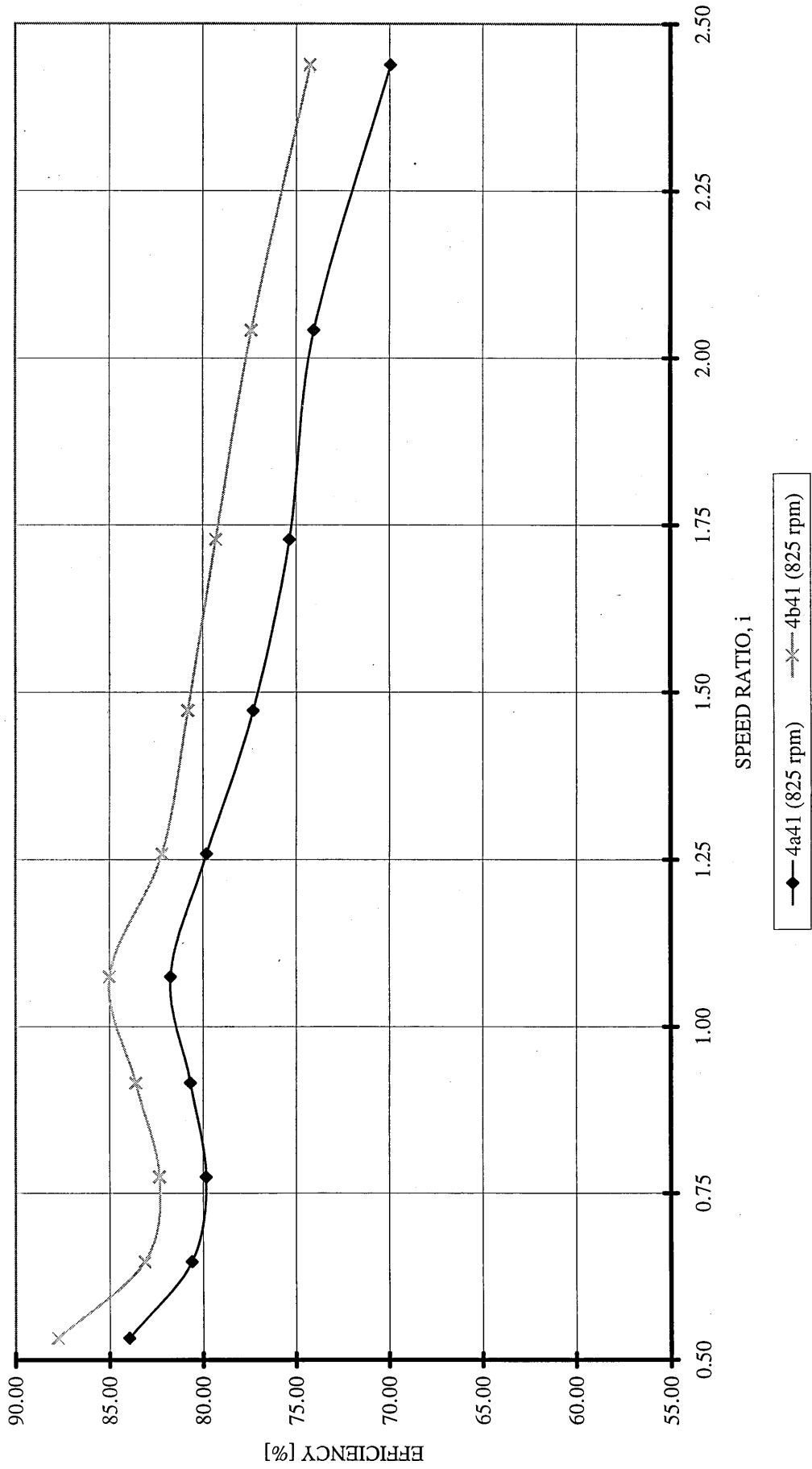


Figure 8.27 Transmission efficiency (Calculated, Constant output torque = 15 N.m) - Expt. 4a41 and 4b41

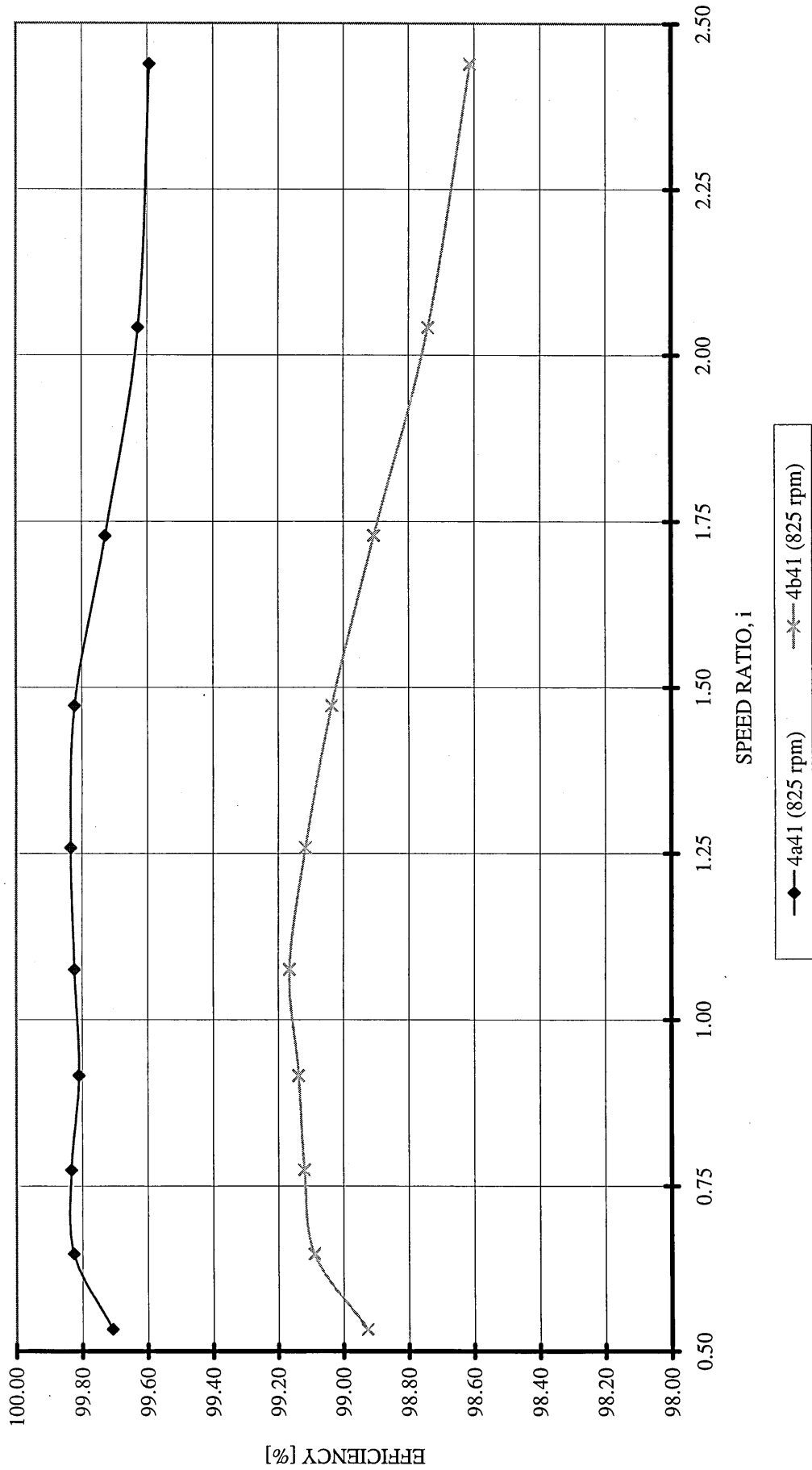


Figure 8.28 Transmission efficiency (Measured, Constant output torque = 15 N.m) - Expt. 4a42 and 4b42

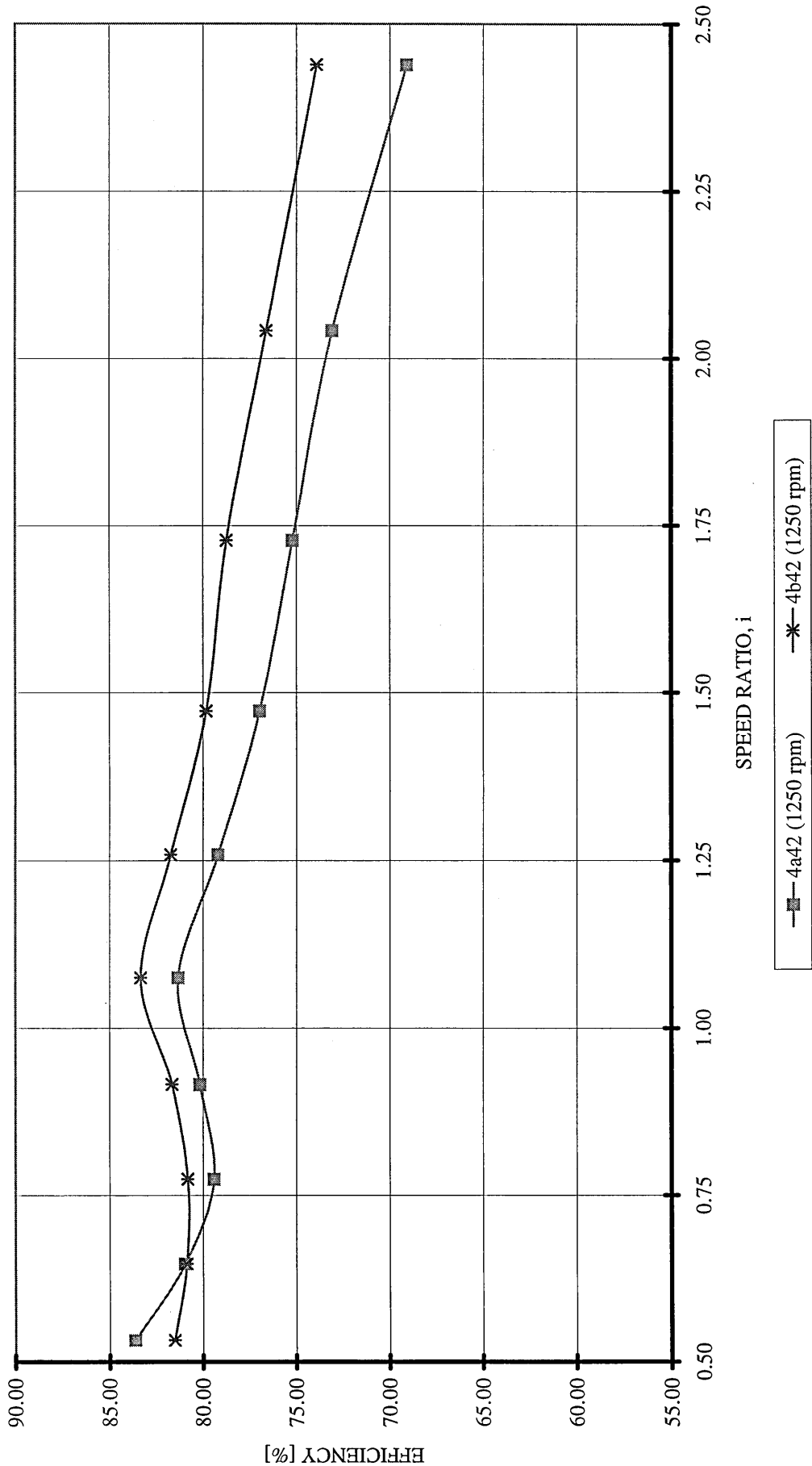


Figure 8.29 Transmission efficiency (Calculated, Constant output torque = 15 N.m) - Expt. 4a42 and 4b42

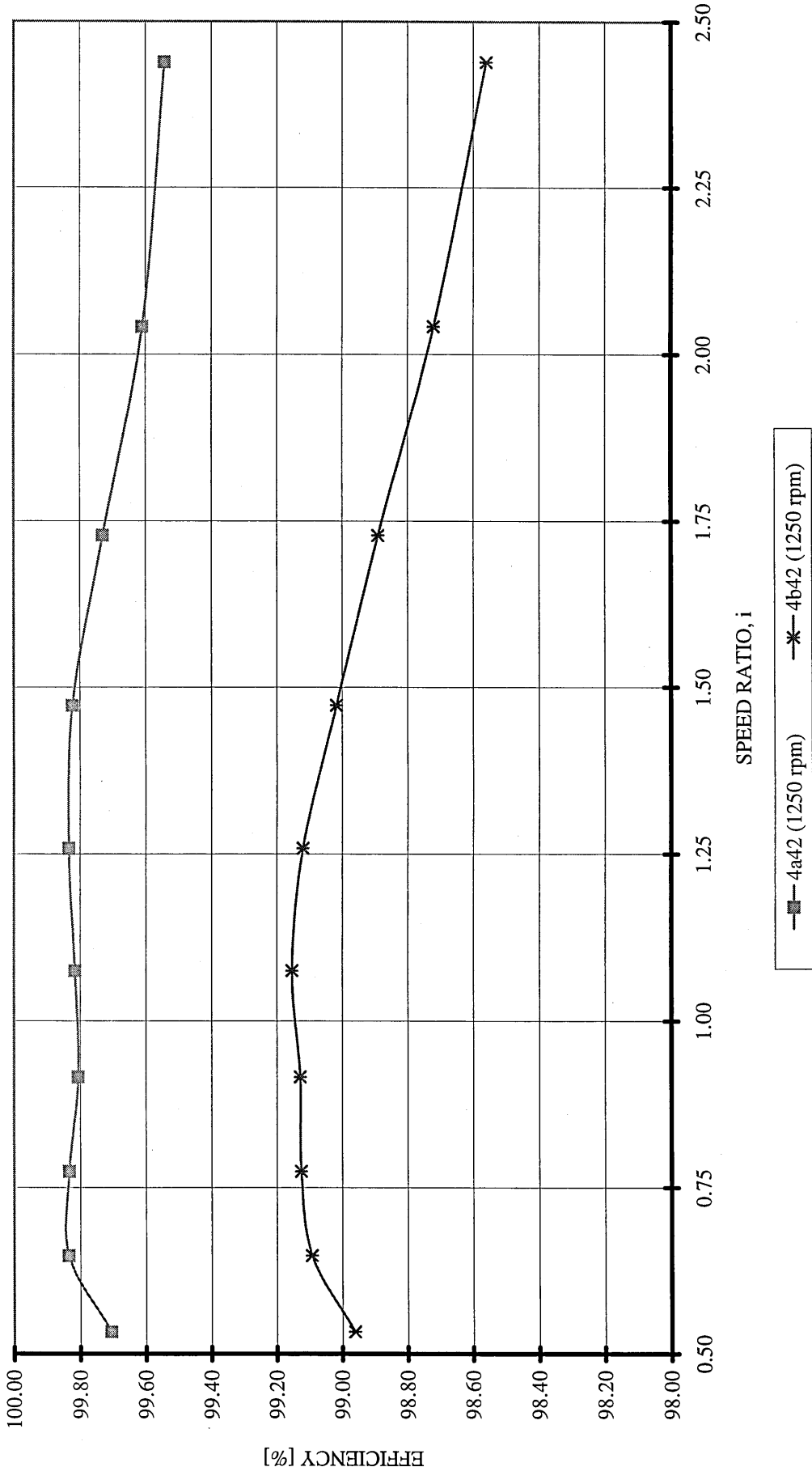


Figure 8.30 Transmission efficiency (Measured, Constant output torque = 15 N.m) - Expt. 4a43 and 4b43

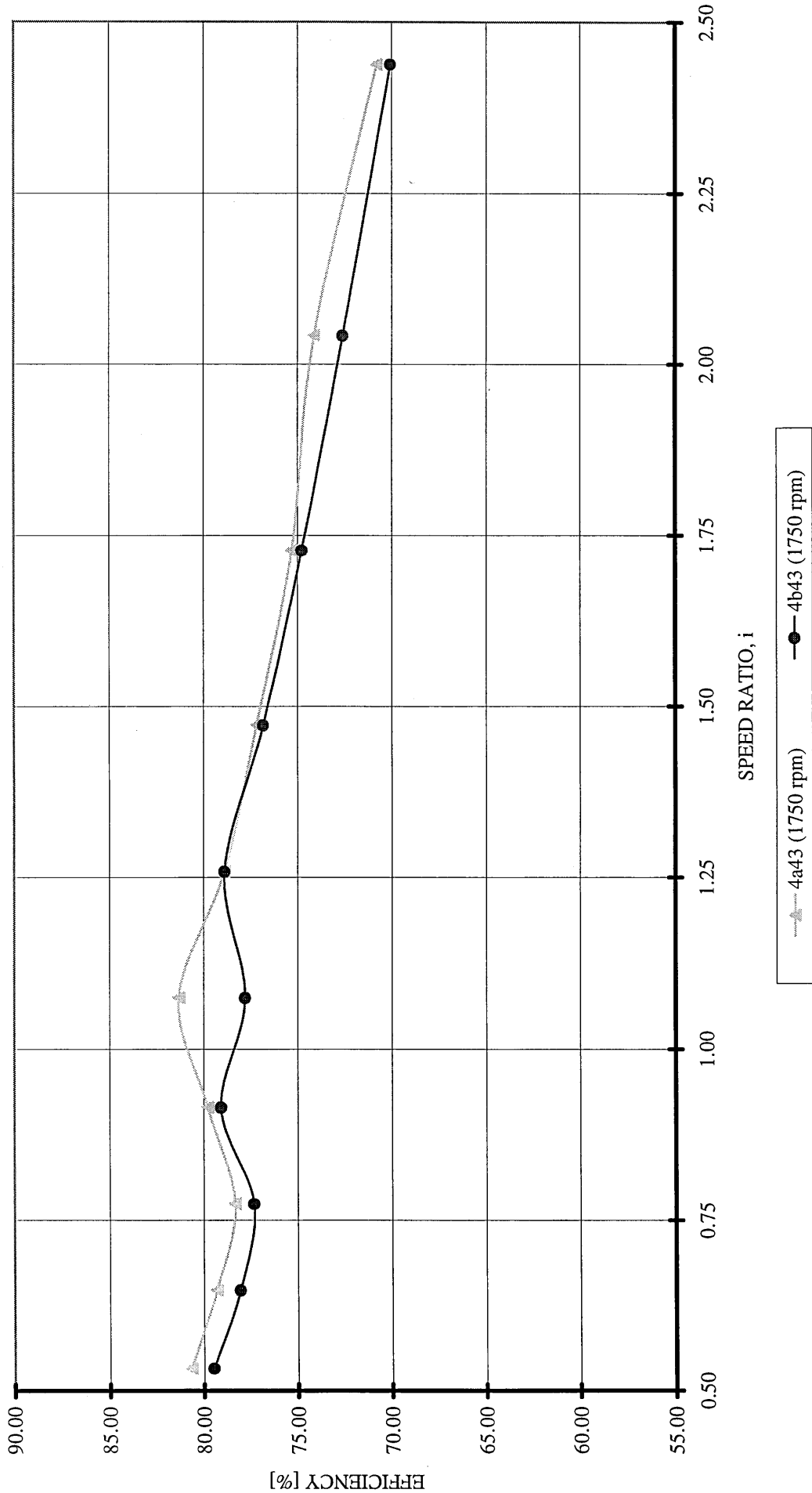
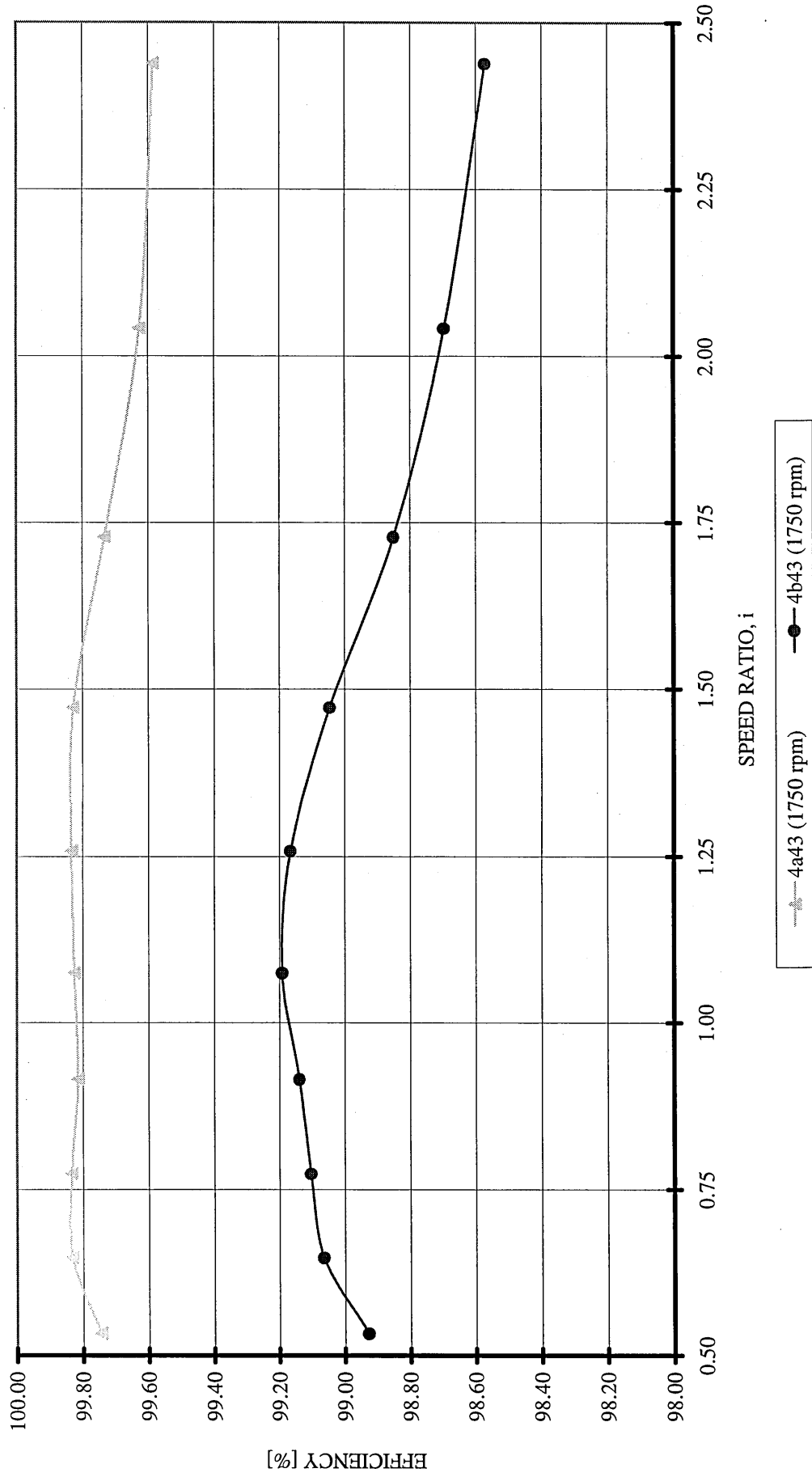


Figure 8.31 Transmission efficiency (Calculated, Constant output torque = 15 N.m) - Expt. 4a43 and 4b43



CHAPTER 9

DISCUSSION

9.1 The Test Rig

The manually controlled test rig performed reasonably well. All the planned experimental work and measurements were successfully carried out. No major problems were encountered through out the experimental work. To ensure repeatability of the experimental work, the author spent a reasonable length of time practising to control the transmission on the test rig. The variable parameters that needed to be controlled and monitored were the primary and secondary pulley speeds, pressures, torque and radii. Before any test was made, the author always tried a few times first to control the system to achieve the required transmission set up conditions. When a reasonable confidence limit had been achieved, only then the actual test results were recorded.

9.2 The Transmission

The transmission with the modified casing, also performed reasonably well. No major problems were encountered through out the experimental work. Dismantling and assembling it back for the purpose of fitting in the spacer at the secondary pulley shaft, was successfully carried out within the expected time of about three quarters of a working day. The viewing window on the transmission box to allow the belt behaviour to be viewed directly with the optical instrumentation while the transmission was in action, functioned reasonably well. This proved to be very helpful because, the belt misalignment movement and position were witnessed while the test was being done. The actual position and movement of the belt, as shown in Figs. 8.4, 8.5, 8.8 and 8.9 could be seen taking place while the test was carried out.

9.3 Validation Of Experimental Pulley Radii Measurements Using The Optical Instrumentation

Figure 8.1 shows that, experimental measurements of the pulley radii using the optical instrumentation agreed reasonably well to the calculated values. The percentage difference between the two, reduced gradually to nearly zero when the primary pulley

radius at its maximum value (the overdrive position). The difference is about 1.48 % at minimum primary pulley radius (underdrive) reducing to about 0.18 % at overdrive position. These results show that, the theoretical model has produced results that represent the actual belt system. The lack of scatter in the experimental points also indicates that the optical measurement technique has proved satisfactory. The difference that occurs between the optical measurements and the calculated values could be due to the fact that, the calculated values were based on unloaded belt length whereas, the measured belt misalignment values were taken when the belt was in tension. Since, practically the belt extends when it is loaded as explained in assumption (i) of the next section, there will always be a slight difference between the measured and the calculated values. The optical instrumentation measurements were made by the author on a flat surface had showed reasonably small scatter with an average standard deviation value of about 0.056 mm. Thus, the experimental errors contributed by the optical method should be very small. However, the main reasons why the author opted to use optical instrumentation are because :-

- i. to avoid any contact with the belt that could incur friction losses while taking measurements and also to avoid extra force being pressed on the bands that could affect the pulley radii,
- ii. to be able to view the geometrical behaviour of the belt, while the test is being carried out and
- iii. it was considered that the transmission lubricant would cause inaccurate measurements and damaged electrical sensors.

9.4 Validation of Belt Misalignment Model

The belt misalignment model is based on the assumption that it will behave exactly like any ordinary rubber V-belt and that classical V-belt theory can be applied to it. Also, it is assumed that the belt behaves ideally, that is:

- i. The belt can be represented by an undeformable inextensible flexible material. Practically, this will not be so because, the belt will extend when it is loaded. Since the model is based on an unloaded belt length hence, the results from the belt misalignment model will not represent the true belt misalignment values.

That is, some small experimental error will be incurred into the model. However, these errors are found to be quite small (previous section) hence, any further refinement of this assumption will be insignificant.

- ii. The belt will enter and exit the pulleys tangentially.

This assumption is probably true, if measurements are to be taken statically while the belt is in tension. However, if the belt is in running condition, it will probably curve in while exit and curve out while entering the pulley grooves [Micklem J. D. et al, 1994a]. When these take place, the belt geometrical dimensions will no longer be in its original assumed conditions. In this work however, the belt misalignment measurements were carried out while the belt was in the static condition. Hence, the results from the belt misalignment model would not be significantly altered by any refinement of this assumption. Furthermore, checks that were made while the transmission was in operating conditions showed quite good agreement with the calibrated results. In fact, by doing so it helps to improve the consistency of the experimental results in using the optical instrumentation as a means of measurement for this work.

- iii. The belt is uniform and continuous.

This is of course not true for the metal pushing V-belt. The belt itself is made up of discrete segments and bands. Thus, it is definitely not uniform and continuous and will affect the belt geometrical dimension. Its free body diagrams are also different from classical rubber V-belt concept. However as explained in assumption (ii), the belt misalignment calibration measurements were carried out statically; thus, this assumption has little or no significant effect to the results obtained by the model. Hence, any fine refinement of this assumption on the model will be insignificant.

- iv. There is no power loss.

Practically this will not occur. It is a well known phenomena that friction losses are inevitable in any mechanical system, and it will definitely incur errors in one form or another to any theoretical models if it is not included. As

for the metal pushing V-belt CVT, there are quite a number of sources of power losses due to the nature of its design. However, as explained in assumptions (ii) and (iii), this assumption should also have little or negligible effect to the model because measurements were taken statically. Hence, no further refinement should be required for this assumption.

Fig. 8.2 shows the belt axial movement as the speed ratio varies from overdrive (0.5329 speed ratio) to underdrive position (2.4385 speed ratio) in graphical form. The difference between these two curves represents the magnitude of the belt misalignment as shown in Fig. 8.3. As predicted (Chapter 3), the belt misalignment geometrical pattern changes its gradient twice over its speed ratio range. These can be seen from Fig. 8.2 where the two curves cross over each other twice. That is, at speed ratio of about 0.70 and 1.40 where the belt misalignment values are zero. These two speed ratio values, therefore represent the position where the two ideal straight planes take place. Assuming that the initially assembled ideal straight plane take place at the former, which corresponds to a primary pulley radius of about 42.25 mm (App. 1.15) and secondary pulley radius of about 59.5 mm (App. 1.15) show that, the calculated values agree reasonably well. The calculated values are shown as dotted lines in all the figures.

Fig. 8.3 shows the comparison between the measured belt misalignment and the calculated values. Both the measured and calculated misalignment curves of Fig. 8.3 were derived from the values in Fig. 8.2. The results show that, the calculated values from the developed model (equation 3.2.11) agree reasonably well with the measured belt misalignment values. The model predicts that, the initially assembled ideal straight plane takes place at speed ratio $i = 0.7024$ (2.28 % difference), while the second ideal straight plane will take place at the reciprocal of the initially assembled ideal straight plane speed ratio position, that is at speed ratio $i = 1.4237$ (3.07 % difference). The maximum belt misalignment of the transmission as predicted by the model, takes place at the transmission underdrive position (at speed ratio, $i = 2.4751$). The maximum measured value is found to be - 0.70 mm while the model predicts a maximum value of - 0.6459 mm (about 7.73 % smaller).

One interesting pattern that can be seen from the calculated belt misalignment values is that, on average they are smaller than the measured results (Fig. 8.3). This phenomena could be due to the fact that the belt does extend when it is underload, which is not assumed in the model, as explained in assumption (i) at the beginning of this section. From the model calculations, it shows that the belt is always in tension through out the operating range. Micklem J. D. 1994, also showed in his calculations that the belt extended while in operation. In this work, the maximum calculated belt tension is found to be about 2900 N . Micklem J. D. et al 1994a said that the maximum value of their belt tension was approximately 4000 N . Thus 2900 N should be reasonable for this work because the maximum input torque of this work is only about 35 $N.m$. A quick check on the model calculations show that, as the belt length increases the belt misalignment values decrease and vice versa. These, support the author's argument on the reason why the calculated belt misalignment values are smaller than the measured ones. That is, the belt was assumed inextensible by the model. In actual fact this is not the case and the belt was measured unloaded. Hence, the model's calculated belt misalignment values should always be smaller than the measured ones.

Therefore, for the model to predict more realistic belt misalignment values, it has to be able to calculate the actual belt length under operating condition as well. This, the author leaves for future work. If however this information is available, it should be able to be incorporated into the model easily by just substituting the belt length as a variable parameter instead of a constant. Despite all this, the model as mentioned earlier predicts the belt geometrical behaviour reasonably well. This can clearly be seen from Fig. 8.4 and 8.5, where Fig. 8.4 shows the measured results and Fig. 8.5 shows the calculated version. These figures show the belt position as viewed into the paper of Fig. 5.7. Take Fig. 8.4 for example, from top to bottom it shows that the belt was initially misaligned with positive gradient at speed ratio of about 0.5329, and as the speed ratio increases, its gradient gradually decreases to zero. The gradient finally reduces to zero at speed ratio approximately equal to 0.70, where the initially assembled ideal straight plane (not shown) takes place. From there on, the gradient

gradually increases again, but in the opposite direction. The gradient continues to increase until the speed ratio is about 0.95, which corresponds to the peak of the curve of Fig. 8.3. The gradient then begins to decrease again to zero until it reaches the second ideal straight plane speed ratio value at approximately 1.40. From here the gradient begins to increase again with positive gradient and reaches its maximum value at the underdrive position. As a summary, from underdrive to overdrive, the belt started with positive belt misalignment gradient with a belt misalignment value of about 0.3 *mm*; it then reduces to zero at the initially assembled ideal straight plane speed ratio; then it begins to increase again in the opposite direction until the belt misalignment reaches a maximum value of about 0.16 *mm*; then it reduces back to zero at the second ideal straight plane speed ratio; finally the belt misalignment gradient increases again positively until the belt misalignment reaches its maximum value of about 0.7 *mm* at the underdrive position.

From Fig. 8.4 and 8.5, it can clearly be seen that the belt moves quite a distance axially along its shafts as the speed ratio increases from overdrive to underdrive position. From the measured figure (Fig. 8.4), it shows that at the primary pulley shaft, the belt moves axially along its shaft a total distance of about 7.30 *mm*, while the model predicts a total distance of 7.25 *mm* movement which is about 0.68 % shorter. At the secondary pulley shaft, the measured figures show that the belt moves axially along its shaft a total distance of about 6.90 *mm*; while the model predicts a total distance of 6.88 *mm*, which is about 0.29 % shorter. All these indicate that the belt misalignment model produces results that represent reasonably well the actual belt misalignment values.

Information from these two figures, should be of great use in designing the two pulley shafts, with regards to bending stresses and vibrations. When belt misalignment is large, this will tend to cause an axial load on the pulleys causing them to tilt on shafts and this may be an explanation why the transmission oil leaking between the two pulley halves and vibration of the transmission were greater at certain speed ratios, in addition to the increase of angular rotation which the pulleys are experiencing. This in turn produced a reduction in the transmission hydraulic pressure and hence, the

requirement of readjusting the manually controlled experimental test rig to maintain steady state conditions.

Fig. 8.6 to Fig. 8.9 show similar results to the previous experiments but with the 2.00 mm offset at the secondary pulley. By doing this, the initially assembled ideal straight plane will be shifted by -2.00 mm and the maximum belt misalignment geometrical behaviour will all be shifted by the same amount. Since, the maximum positive belt misalignment value is about 0.16 mm (Fig. 8.3) hence, there is no ideal straight plane condition for this part of the experiment. It is evident from these figures that what is expected did take place. The initially assembled ideal straight plane had shifted by -2.00 mm and the rest of the graphs also shifted by the same amount, both the measured and the calculated values agree reasonably well. Therefore, in this situation, the belt is always in misalignment conditions at all the transmission speed ratio range (Fig. 8.8 and 8.9 depict this situation clearly). Comparison of misalignment values with the additional spacer fitted show that the minimum measured belt misalignment at speed ratio $i = 0.95$ is now -1.84 mm as compared to 0.16 mm in the original condition, while the maximum measured misalignment at $i = 2.4385$ is -2.70 mm as compared to -0.70 mm. The calculation in the model predict a minimum misalignment value of -1.86 mm as compared to 0.14 mm at speed ratio $i = 1.00$ and a maximum misalignment of -2.65 mm as compared to -0.65 mm at speed ratio $i = 2.4385$. These, therefore justify the consistency and accuracy of the belt misalignment model in predicting the actual belt misalignment values and behaviour.

9.5 Validation Of The Efficiency Model Due to Belt Misalignment

9.5.1 Introduction

The simplifying assumptions used in this model are:

- (i) The pulley cone surfaces are relatively more rigid than the segments, hence for this analysis, only the segment elasticity is considered.

It is evident that from their physical size, the pulley cone surfaces are relatively more rigid than the segments. Checks had been made by punching the side of one of the pulley cone and it was found that, there were hardly any marks that

resulted from these punches i.e. the surface is very hard. Hence, no further refinement should be needed for this assumption.

- (ii) The segments will be compressed uniformly and equally at both ends.

The pulleys had been designed in such a way that the force flow while compressing the segments is taken up equally by both halves of the pulleys. Take for example the primary pulley system as shown in Fig. 1.8; as the pressure increases, both the pulley cones compress the segments; the force flow begins from the pressure force inside the cylinder of the adjustable half of the pulley; the pressure force then pushes the adjustable half of the pulley; the pulley in turn pushes the segments; the segments then push the fixed half of the pulley cone; the fixed half through its shaft then pulls in the end nut near the oil pump; in return, the reaction from the pressure force inside the adjustable half of the pulley pushes its cylinder cover against the washer, the washer then pushes the inner ring of the bearing which in turn pushes the end nut; thus, leaving the pulley fixed-half shaft in tension and hence, the segments will be compressed equally at both ends. A similar force flow is taking place on the secondary pulley system because its design principle is the same. Hence, no further refinement should also be needed for this assumption.

- (iii) There is no belt slip

One would say that, due to the nature of Van Doorne metal pushing V-belt CVT designed itself, the risk of slipping taking place is high. Furthermore, the metal belt is fully lubricated with transmission oil which is a favourable condition for belt slip. But in order to provide a satisfactory drive the clamping of the pulley cones by the hydraulic pressure must minimise the slip between the cones and segments. Therefore, this simplified model assumes no belt slip.

- (iv) The belt is uniform and continuous.

The belt is definitely not uniform and continuous as has been explained in assumption (iii) of the previous section (section 9.4). Since in this case, the experiments were carried out while the transmission is in operating condition,

as compared to the static case carried out in section 9.4 hence, there are some effects on the calculated model. Many attempts have been made by quite a number of researchers in trying to represent and analyse the forces of the metal pushing V-belt CVT (Chapter 2). However due to the complexity of its design, most of them opted to use simplified models to represent the belt mechanics. The author has to admit that, it is not an easy task to model the metal pushing V-belt CVT mechanics. Even to come out with reasonable free body diagrams to represent the belt forces proved to be very complex. Quite a number of conditions had to be considered. To name a few:

- a. Properties of the material itself, such as mass, elasticity, uniformity etc.
- b. Coulomb friction, elastohydrodynamic lubrication or probably something else as the media between the relative surfaces.
- c. If Coulomb friction, it could be either static or kinetic condition or probably a combination of both.
- d. If elastohydrodynamic lubrication, a further number of other properties are needed to be considered, such as viscosity, temperature, surface finish, lubricant thickness, components relative speeds etc.
- e. Relative motion of almost all components of the metal pushing V-belt CVT, such as between segments, bands, segments and bands, segments and pulleys, bands and pulleys etc.
- f. Misalignment of the belt.
- g. Belt behaviour during exit from and entry into the pulley groves.

These are just a few and there are probably many more that need to be considered. Even to just consider the relative motions between the belt components themselves are already quite a complex exercise to undertake. All these, highlight the complexity in trying to model the metal pushing V-belt CVT mechanics. The author himself, had spent a major part of this work trying to come out with a reasonable model that could simulate the belt performance with regards to its belt misalignment problems. Hence, this assumption does need some refinement in order to achieve a more realistic prediction of the

transmission efficiency with regards to the belt misalignment effects. This, the author reserves for future improvement of this work.

- (v) The pulley clamping forces remains relatively constant during operation.
As had been mentioned in section 9.1, when a reasonable confidence limit had been achieved by the author, only then the actual test was carried out. In addition to this, the use of the data acquisition system (section 7.7) had allowed thousands sets of measurements to be taken within a short period of time which could be averaged to reduce the effects of any mechanical and electrical noises. Thus, this assumption could be applied in this work and no further refinement of the assumption should be required because the necessary precautions and conditions had been taken and carried out.
- (vi) The belt leaves and enters the pulleys tangentially.
Micklem J. D. et al, 1994a, showed in their work that the belt did actually curved slightly while entering and leaving the pulley cones. In this work however as this curve is very small, the author assumed that this assumption for simplicity purposes.
- (vii) The bands and segments travelling along the compression straight will remain relatively plain over the length X' (refer Fig. 5.2). That is, they do not have a radius of curvature. Thus, friction forces between the bands and the segments shoulders along the compression straight are considered negligible.
This is to be in consistence with the previous assumption. If the belt curves in or out while leaving or entering the pulleys, then this assumption needs some further refinement which includes centrifugal forces of the belt and friction forces between the bands and the segment shoulders.
- (viii) Coulomb friction exists between all the sliding components.
This has been explained in the discussion section (section 1.4) of Chapter 1.
- (ix) Only steady state conditions are considered.

As the transmission test facility was designed for simplicity without a control system to vary loads, speeds and ratios, only steady state conditions were considered.

9.5.2 Belt Slip

Fig. 8.12 to 8.15 show the plots of speed ratios that were measured using the two experimental methods available on the test rig. That is, by using pulley radii and speeds. Pulley radii measurements were taken using the optical instrumentation while pulley speeds were measured using the speed meters. All the figures show that, the maximum difference between the two measured values took place at low gear (maximum speed ratio) or underdrive position with the one based on pulley speeds greater than the one based on pulley radii. The difference reduces gradually to about zero as the speed ratio decreases. All these indicate that, maximum overall belt slip always takes place when the transmission is in low gear position and reduces to almost zero as it approaches the overdrive position, irrespective of the input and output torque and belt misalignment values. Therefore, the assumption made by this model regarding belt slip as mentioned in the previous section needs to be modified to include this phenomena which the author leaves for future work.

9.5.3 Transmission Efficiency

For the constant input torque case (Fig. 8.16 to 8.23):

- i. Expt. 4a11 and 4b11 (Fig. 8.16 - measured and Fig. 8.17 - calculated) for constant underdrive condition ($i \approx 2.4385$).

Overall, the measured results for both the standard transmission conditions (Expt. 4a11) and with the 2.00 mm additional spacer (Expt. 4b11) set-up show that, as the speed increases the transmission efficiency stay relatively the same with some indications of unstability probably due to experimental errors. The average difference of efficiency drop due to belt misalignment, was about 1 %. The calculated values, predict the efficiency trends reasonably well for both cases. Although the absolute values are not comparable, the predicted average efficiency drops about 0.8 % agrees quite well with the measured values.

- ii. Expt. 4a12 and 4b12 (Fig. 8.16 - measured and Fig. 8.17 - calculated) for constant overdrive condition ($i \approx 0.5329$).

Rather interesting results were seen to take place for this condition. That is, by increasing belt misalignment the transmission efficiency actually increases, instead of reducing it which is opposite to the results predicted by the model. The efficiency trends are flatter than the previous case and the efficiency patterns for the two cases are quite distinct with an average increase of about 1 %; while the calculated values predict an average drop in efficiency of about 1 %, instead; no explanation could be given for this. However, the model predicts the transmission efficiency trends reasonably well.

- iii. Expt. 4a31, 4b31, 4a32, 4b32, 4a33 and 4b33 (Fig. 8.18 to 8.23; even numbers - measured, odd numbers - calculated).

Almost similar results to the previous two results were seen at the underdrive and overdrive conditions. Increasing belt misalignment increases the transmission efficiency slightly at overdrive condition and vice versa at underdrive, and as the input speed increases, its effects tend to decrease slightly. For the 825 and 1250 *rpm* input speed cases, the results show that as the speed ratio increases, the efficiency trends show a transition region of the curves crossing each other; that is increasing the belt misalignment increases the transmission efficiency at the overdrive position and decreases it at the underdrive, where in between these two speed ratios a transition region from a favourable condition to an unfavourable one takes place. As for the 1750 *rpm* input speed, the results show that through out the speed ratio range, the transmission efficiency gradually drops with maximum value at the mid-range, minimum at the over and underdrive positions, with a slightly greater loss at the underdrive. The calculated values however, only predict the overall efficiency trends reasonably well but not their absolute values.

For the constant output torque case (Fig. 8.24 to 8.31):

- i. Expt. 4a21, 4b21, 4a22 and 4b22 (Fig. 8.24 - measured and Fig. 8.25 - calculated) for both over and underdrive conditions.

In general the results show that, offsetting the belt misalignment by 2.00 mm decreases the transmission efficiency by about 2 to 5 %. The results also show that the transmission efficiency is greater at overdrive position than underdrive. These agree reasonably well with the model because at underdrive, the belt misalignment value is greater than overdrive (Fig. 8.3 and 8.7) thus, the transmission efficiency should be greater at the overdrive compared to underdrive condition. The results also show that the efficiency trends for both under and overdrive conditions are increasing slightly as the input speed increases. The model predicts the efficiency trends reasonably well but its absolute values are not comparable; however, the model predicted values of efficiency drop of about 0.7 % for the overdrive case and about 1 % for the underdrive condition while the measured results are about 3 % for both cases.

- ii. Expt. 4a41, 4b41, 4a42, 4b42, 4a43 and 4b43 (Fig. 8.26 to 8.31; even numbers - measured, odd numbers - calculated).

The results show similar trends as in (i) that is, increasing belt misalignment in general reduces the transmission efficiency slightly, and the overdrive efficiency is better than the underdrive. These, agree reasonably well with the model. The results also show that, the efficiency drop is almost constant through out the whole speed ratio range but its values gradually decrease as the input speed increases. At the higher input speed it shows that, the effect of increasing belt misalignment is very small and sometimes almost insignificant. The model predicts the efficiency trends reasonably well except that it has a slight disagreement towards the overdrive conditions. The model however, did not predict the absolute values of the transmission efficiency correctly.

Overall, the measured results show that the model which was based on belt misalignment effects predicts the transmission efficiency trends reasonably well but not their absolute values. The hypotheses that the transmission power loss is associated with the magnitude of the belt misalignment values seems to agree only partially. What the author would expect to see from the work is that, the transmission efficiency trend should reach its maximum value at the speed ratio of the initially

assembled ideal straight plane and at the second ideal straight plane for the nominal pulley set-up position where, the belt misalignment values are zero. In this case they are expected to take place at speed ratios of about 0.7024 and 1.4237. For the 2.00 mm belt misalignment offset case, there should only be one maximum values and it should take place at speed ratio of 0.95 the peak of the curve of Fig. 8.7, where the belt misalignment value is a minimum. Unfortunately, all these expectation did not take place.

All these results, suggested that there were also other main contributors of power losses on the transmission. In this work, the two major contributions were vibration and belt slip which the model did not consider. With regards to the vibration, the work had witnessed that there were considerable amounts of vibrations at certain speed ratios and input speeds. These might have had some relation with the belt misalignment phenomenon. It is evident from Fig. 8.4 and 8.8 that, vibrations could result from the belt misalignment phenomenon. Examination of Fig. 8.12 to 8.15 clearly show that belt slip did take place. Since vibration and belt slipping lead to sliding between various components, more energy will be loss in terms of heat due to the sliding friction. If the behaviour of these vibrations and belt slipping could be studied and their magnitudes and frequencies could be predicted with respect to the transmission speed ratio, then the work in this thesis should be able to be extended to include these effects. These, could then help to explain or validate the outcomes from the work that has been carried out in this thesis. Consequently, the hypothesis put forward in this thesis could still hold and the author leaves for future work.

The measured results in general had shown that, the transmission efficiencies were quite low. On average, the maximum transmission efficiency witnessed in this work was about 85 %, which is slightly lower than most published work in this field. The most recent published work on the Van Doorne metal pushing V-belt CVT by Rine Pelders in the proceedings of the International Seminar on Advanced Vehicle Transmissions and Powertrain Management, London 1997 regarding the transmission overall efficiency reveals that, the transmission efficiency is highly dependent on the clamping force or what he called safety factor reproduced here as Fig. 9.1. He quoted

that, the minimum value for the pulley clamping pressure should be about 5 *bar* (72 *psi*). The constant secondary pulley pressure used in this work (280 *psi* or 19.31 *bar*) however, was far above this minimum value. That is, a safety factor of about 3.86 which means that the overall transmission efficiency should be in the region of about 73 %. Thus the measured transmission efficiency of about 85 % in this work agrees with Rine Pelders results. Checks from the test rig, confirmed this. Fig. 9.2 and 9.3 shows the efficiency comparison between the one running at a lower clamping secondary pulley pressure of about 8.96 *bar* (130 *psi*, a safety factor of about 1.79) with the one running at 19.31 *bar* (280 *psi*, a safety factor of about 3.86). Overall, the figures show that there is an increase in the transmission efficiency of about 5 % which thus confirms the results obtained in this work and hence, explained why the measured efficiency values in this work were quite low. Rine Pelders 1997, also revealed the overall efficiency of the Van Doorne metal pushing V-belt CVT reproduced here as Fig. 9.4 which further justifies the measurements made in this study.

9.5.4 Theoretical Anticipated Changes With Regards To Input Variables

In this section, theoretical investigation results comparable to experiments Expt. 4a32 (nominal set up) and 4b32 (with 2.00 *mm* spacer added) that is, for constant input torque case with input speed of 1250 *rpm* are discussed. This is to highlight the theoretical anticipated changes predicted by this model for varying the input variables such as coefficient of friction and secondary pulley pressure. Also, in this section experimental measuring errors are shown and discussed.

Appendix 1.16 shows the anticipated efficiency changes predicted by the model if the coefficient friction value is decreased to 0.08 or increased to 0.13. The results show that increasing coefficient friction decreases the transmission efficiency and vice versa. This is because increasing the coefficient of friction value in turn increases the friction force between the belt components interfacial surfaces. Hence, this increases the transmission power loss due to sliding between the segment back and front surfaces along the compression straight, and between the segment sides and the pulley cone surfaces during entry into and exit from the pulleys. The converse take place if

the coefficient friction value is decreased. These losses should increase as the belt misalignment increases and these can be seen from the three bottom curves of App. 1.16 where extra 2.00 *mm* belt misalignment was introduced.

Appendix 1.17 shows the anticipated efficiency changes predicted by the model if the secondary pulley pressure is reduced from 280 *psi* (19.31 *bar*) to 75 *psi* (5 *bar*). The results show that decreasing the secondary pulley pressure increases the transmission efficiency. However, a rather smaller effect from that expected was seen from this exercise. This indicates that some refinement of the model need to be carried out which the author leaves for future work.

The transducer used to measure the pulley hydraulic pressure was from Gems Sensors, TransInstrumentation Division, Basingstoke type 2000GGB4001A30A. The specification of this obtained from the manufacturer states that its static error band is $\pm 0.25\%$ span typical (of 600 *bar*). This means that the errors for measuring the pressure was within plus and minus 1.5 *bar* of the measured values.

Appendix 1.18 shows the upper and lower limits (standard deviations values) bands for the transmission efficiency values. It is evident from the results that greater error spans took place at the overdrive condition and reduces gradually as it approaches the underdrive condition. These could be due to the vibration effects as the transmission speed increases which needs to be looked into in future work.

Appendix 1.19 shows the data points collected at 200 Hz over 10 seconds time period for Expt. 4a32 for speed ratio 1.0753. These two thousands data points were averaged to determine the nominal efficiency. The graph was quite 'spiky' in nature. However, it does show a general straight line trend which the author feels is experimentally acceptable.

As for the error due to thermal effect, measurements were only taken after the transmission has been kept running for about half an hour. This was to ensure that the transmission fluid temperature had stabilised and thus the coefficient of friction

should remain reasonably constant through out the experiment. Furthermore, the test rig had been equipped with cooling system that helps to maintain the transmission fluid temperature reasonably stable.

An audit that was made on the test rig mechanical losses shows that its bearings losses are quite small. These are the bearings that support the pulley shafts which are situated at each side of the pulleys (Fig. 1.8 and Appendix 1.20). Appendix 1.20 shows the power loss calculation due to these bearing frictional losses and their effect towards the transmission efficiency with regards to belt misalignment for Expt. 4a32 (nominal set up) at zero belt misalignment. From the calculation it shows that the power loss is about 15.58 *watts* (about 0.73 % power loss). The efficiency loss due to the bearings of 0.73 % is small compared with the total measured losses of about 19 % (experimental efficiency for Expt. 4a32 at speed ratio 1.40 was about 81 %). Therefore the bearing losses are less than 4 % of the total measured losses. As for the effect of belt misalignment towards the shaft axial load, the calculation shows that even for the maximum belt misalignment case, its value is quite small (about 1.93 % of the radial load). The increase in bearing load due to misalignment is extremely small and therefore the increase in frictional losses has been neglected.

From all these calculations it shows that although the measurements of power loss and efficiency included the bearing losses, these bearing losses are negligible compared to the belt losses.

9.6 Proposal For Further Work

In this section, proposals for further work on various sections of the study presented in this thesis are discussed.

9.6.1 The Test Rig And Transmission

The work in this thesis has revealed that:

- i. It will definitely be much better if the transmission could be controlled automatically by using computers to control the speed ratios. This could probably shortened the research time and widen the field of research and

development work that could be carried out on the transmission. Also, this will enable a better quality of research work to be completed. However, to design and fabricate an automatically computer controlled test rig for the Van Doorne metal pushing V-belt CVT was beyond the scope of this project. It could probably be proposed as one for the future work on this test rig.

- ii. The viewing windows have proved to be very useful in this work. However, if a full scale testing on the transmission performance is to be done, then it is advisable that a total modification of the window to be carried out.
- iii. At certain speed ratios and input speeds, the transmission vibrates considerably, thus it is suggested that some means of bushing or suspension to be installed at the appropriate section of the transmission box to absorb these vibrations. This is because, as has been explained at the end of the previous section, vibrations might have played a major part in contributing power loss which was not included in this thesis. Thus, if these vibrations could be suppressed or reduced, it might to a certain extent help to clarify the queries raised from this work. As has been mentioned in the previous section, it should also be very useful if the behaviour of these vibrations could be studied as a proposal for future work on the test rig because, its information could help in improving the design and performance of the transmission.
- v. The experimental errors contributed by the optical instrumentation as has been mentioned in section 9.3 were quite small. However, the author believes that these errors could be further reduced to nearly zero by introducing optical lenses that have splitting image focusing facilities, such as those used in the 'single lens reflex' camera. This, will then eliminate totally any focusing and parallax problems with the current optical instrumentation. A camera or video camera that is linked to a computer screen could probably be a better alternative because, the belt in action could then be captured by the camera or video to further study the belt mechanical and geometrical behaviour in greater depth. The splashing of the transmission oil could also be seen in action and its

viscous characteristics could probably be studied to further enhanced the findings of Micklem J. D. et al 1994b.

- vi. The capacity of the current electrical motor and pump were not sufficient to carry out a full test on the transmission. The 1000 *psi* (68.95 *bar*) electrical pump could only deliver up to a maximum of about 300 *psi* to the transmission in this work. Thus, a bigger capacity pump is required if a full scale test is to be carried out.
- vii. The cover plate which replaces the transmission oil pump as has been mentioned in section 7.1 had a small leak. Due to time limitation, further improvement was not carried out. It would be better if this leak could be stopped totally by redesigning the cover plate. This, will enable full scale testing to be carried out on the transmission hydraulic system and hence, widen the test rig scope of research on the transmission.
- vi. Currently, the test rig is only capable of carrying out a test up to a maximum input speed of about 1750 *rpm* and an input torque of about 36 *N.m*. The author would like to propose that these limitation being extended to full scale test on the transmission so that, full scale correlation studies on the transmission belt misalignment problems could be investigated. This will be very helpful because, one can study the full transmission performance behavioural pattern and hence, make a more representative and conclusive conclusion.

9.6.2 The Developed Belt Misalignment Model

- i. The model assumed that the belt could be represented by an undeformable inextensible flexible material. Unfortunately, the belt practically extends when loaded. One way of probably overcoming this, is to calibrate the belt tension against its extension. The calibration results, can then be easily incorporated into the belt misalignment model. However, to measure the belt tension while

the transmission is in operating condition was beyond the scope of this work and could probably be proposed to be looked into as future development work.

- ii. The present work as has been mentioned in the discussion section, did not consider the effect of belt slip. Thus, it is appropriate probably now to try to extend the model to include slipping between segments and pulleys around the angle of lap.
- iii. The present work only covers negative belt misalignment offset and did not cover positive belt misalignment offset effects due to time limitation. It would be useful if this could be done so that correlation studies could be made between the two. Also, it would probably be more conclusive if the extra belt misalignment offsets were introduced.
- iv. Instead of using the rotary concept to measure the pulley radii by the optical instrumentation, it would be better if it could be redesigned to be able to slide parallel to the pulley shafts centre line X (Fig. 7.3) thus, no calibration is required and hence, the values of the pulley radii can be measured directly.
- v. Perhaps the best way of investigating the belt misalignment effects on the Van Doorne metal pushing V-belt CVT is to redesign both the shaft support bearings for one of the pulley so that it could 'float' or slide axially with the minimum friction possible while it is in operation. This, could be done by using sliding bearings at both ends of the shaft, then the force flow plus the tension in the belt will always tend to keep the movable pulley in line with the fixed one. Another way of doing it for experimental purposes is to fix an adjustable end screw at both ends of one of the pulley shafts so that the belt misalignment could be adjusted to the required position. Thus, a direct comparison studies on the effects of belt misalignment can then be done. These however, will require further research and development work because it will also involve a major redesign exercise and the author leaves this for future work.

9.7 Figures

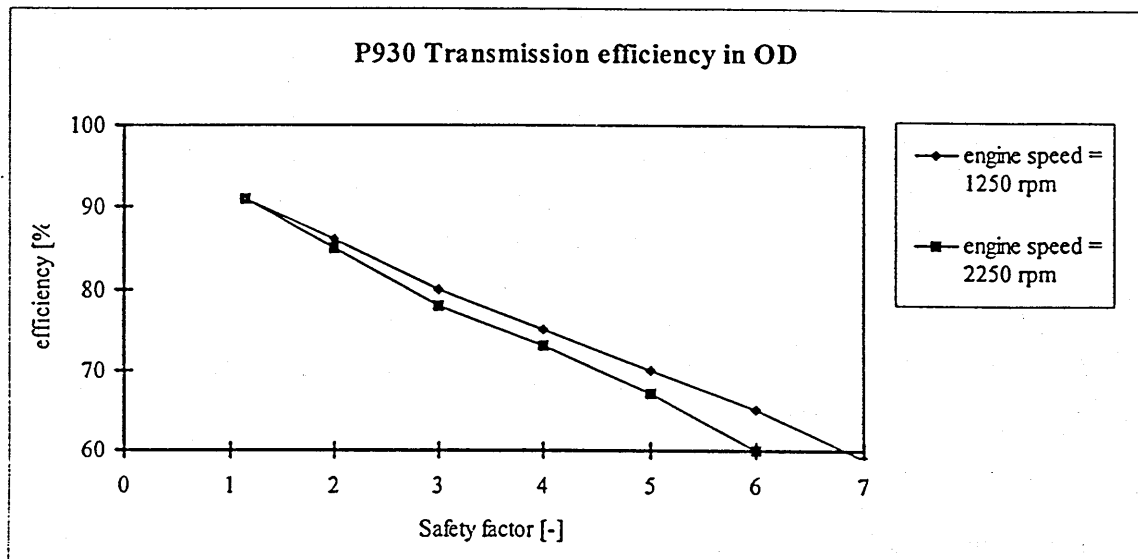


Fig. 9.1 P930 efficiency as a function of safety factor [R. Pelders, in the International Seminar on Advance Vehicle Transmissions and Powertrain Management, 1997]

Fig. 9.2 Transmission efficiency (Measured, $\Delta\epsilon = 0.00$ mm). Constant Output Torque = 15 N.m and Constant Secondary Pulley Pressure = 130 psi (8.96 bar, SF = 1.79)

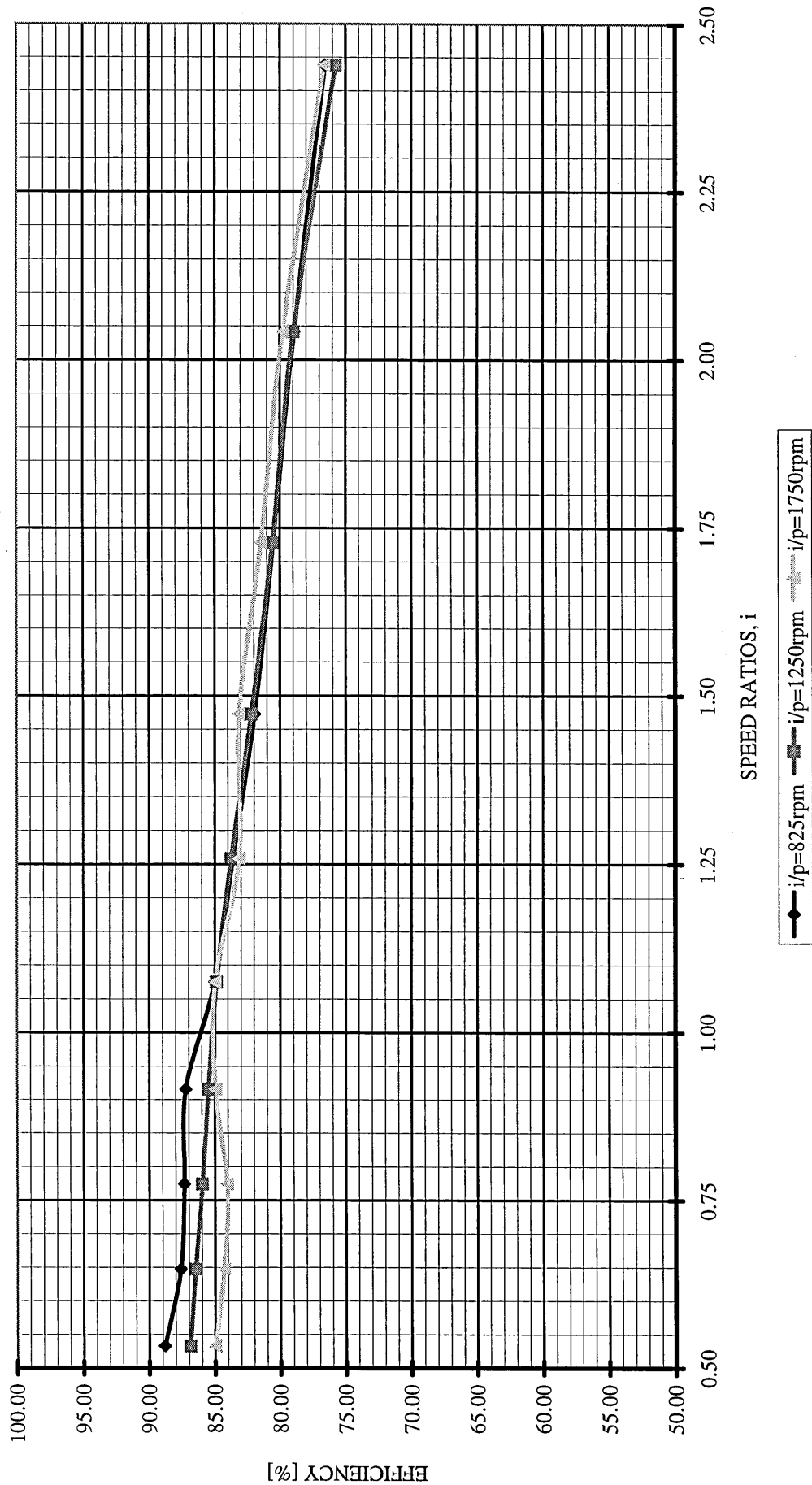
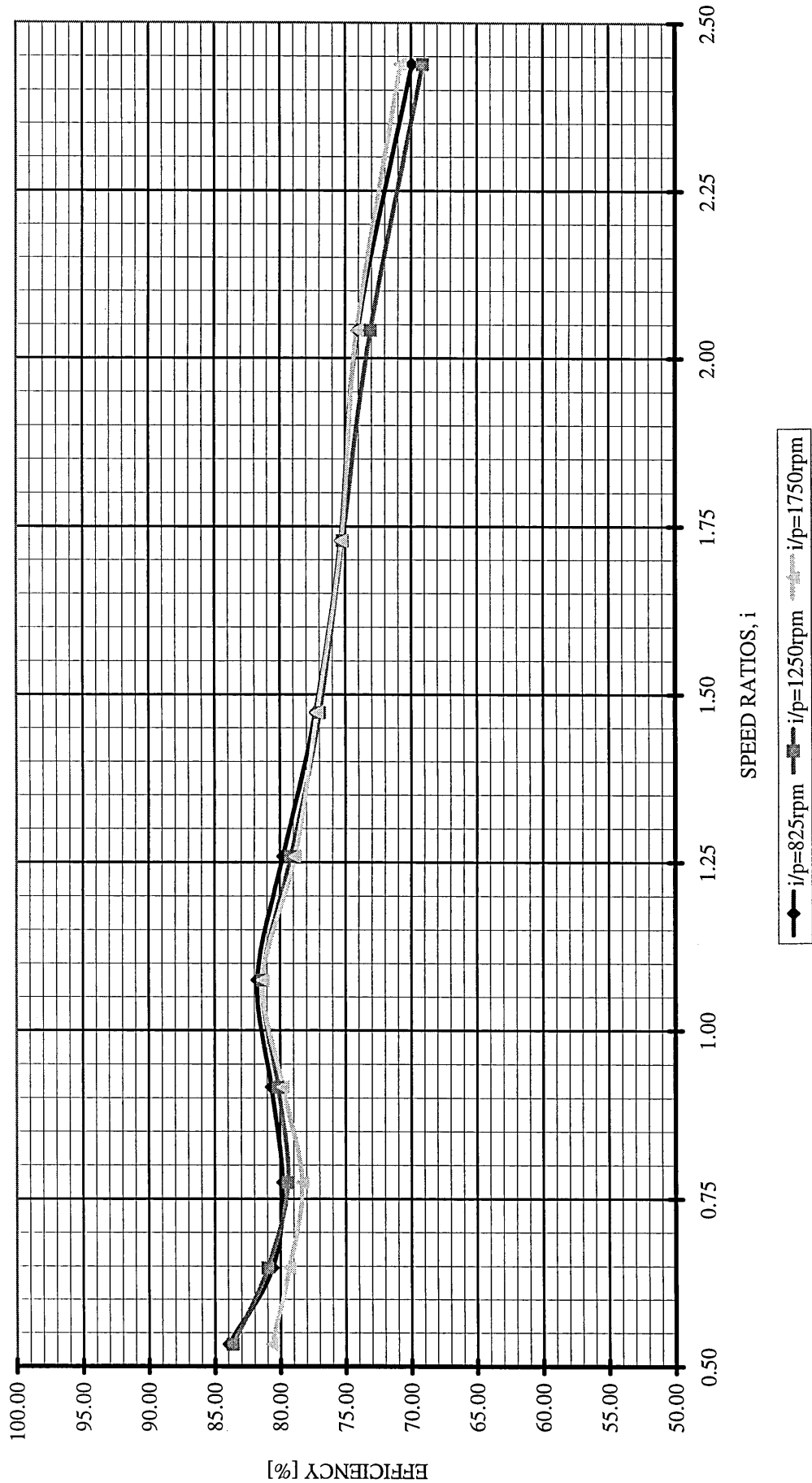


Fig. 9.3 Transmission efficiency (Measured, Δ delta = 0.00 mm). Constant Output Torque = 15 N.m and Secondary Pulley Pressure = 280 psi (19.31 bar, SF = 3.86)



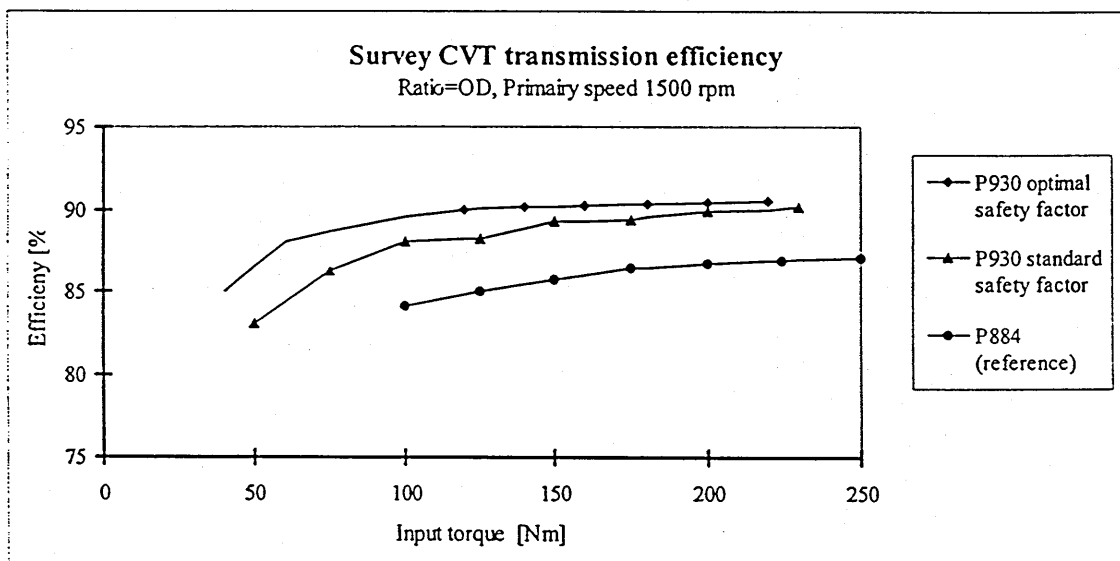


Fig. 9.4 CVT transmission efficiency depending on control strategy [R. Pelders, in the International Seminar on Advance Vehicle Transmissions and Powertrain Management, 1997]

CHAPTER 10

CONCLUSIONS

10.1 Introduction

The study started with the hypothesis that, the Van Doorne metal pushing V-belt CVT belt misalignment problem has significant effects towards the performance of the transmission. From there, efforts have been made to study the actual geometrical behaviour of the belt misalignment. Once, this had been established and the results agree quite well with published paper by Hendriks E. et al 1988, and extremely well with a computer solid modeller; the next step was to try to develop a model that could be used to calculate the efficiency of the transmission with regards to the belt misalignment problems. This was done to validate the experimental work to be carried out solely to study the belt misalignment effects on the transmission efficiency. These proved to be quite difficult, due to the unique and complex design of the Van Doorne metal pushing V-belt CVT itself. This was because, the Van Doorne metal pushing V-belt CVT was made up of discrete components that could move relative to each other. Hence, a simplified model and some simplified concepts from published work were used to develop this model.

Effort was then concentrated in trying to refine the model and design a test rig that could aid this research. Half-way through the test rig development, it was found that it was not feasible due to equipment failure. A second test rig had to be redesigned and built with a totally new concept from the earlier one. The Van Doorne metal pushing V-belt CVT was dismantled and reassembled back into a newly designed gearbox where it was manually controlled. Power was supplied from an electrical a.c. motor and the transmission oil was supplied externally by an electrical motor and pump.

10.2 Belt Misalignment Model

The model predicted the belt misalignment geometrical behavioural pattern reasonably well. However, its prediction values on the average were a fraction smaller than the measured ones. The small percentage difference between the experimental results and the calculated values as has been mentioned in the discussion was due to the model

calculations which were based on unloaded belt length instead of the actual length. Hence, for the model to predict more realistic belt misalignment values, the operating transmission belt length extension has to be incorporated into it.

10.3 Efficiency Model Due To Belt Misalignment

The model has succeeded in predicting the belt efficiency trends reasonably well but not their absolute values. Transmission vibrations and belt slip could be the main reasons for inaccuracies in this model, which was mainly based on sliding friction between the segments due to belt misalignment along the compression straight and friction while leaving the pulley. Vibrations and belt slip will cause more sliding between components and hence, reduce the transmission efficiency which this model did not consider at all.

10.4 Overall

In general, this work has proved to be very useful in the understanding of belt misalignment. The work has shown to a certain extent that:

- i. The transmission efficiency is a function of the belt misalignment, input and output torque, and input speed conditions.
- ii. Increasing the transmission belt misalignment value is not always followed by extra transmission power loss, it is sometimes favourable, sometimes unfavourable and in some cases insignificant. This, the author would consider as an interesting discovery which could be exploited in future development work of the transmission.
- iii. The transmission vibrations were quite apparent at certain speed ratios which might have had an effect on the transmission efficiency.
- iv. Belt misalignment might have an effect on the transmission vibration.
- v. Belt slip did take place and it is not a function of belt misalignment.

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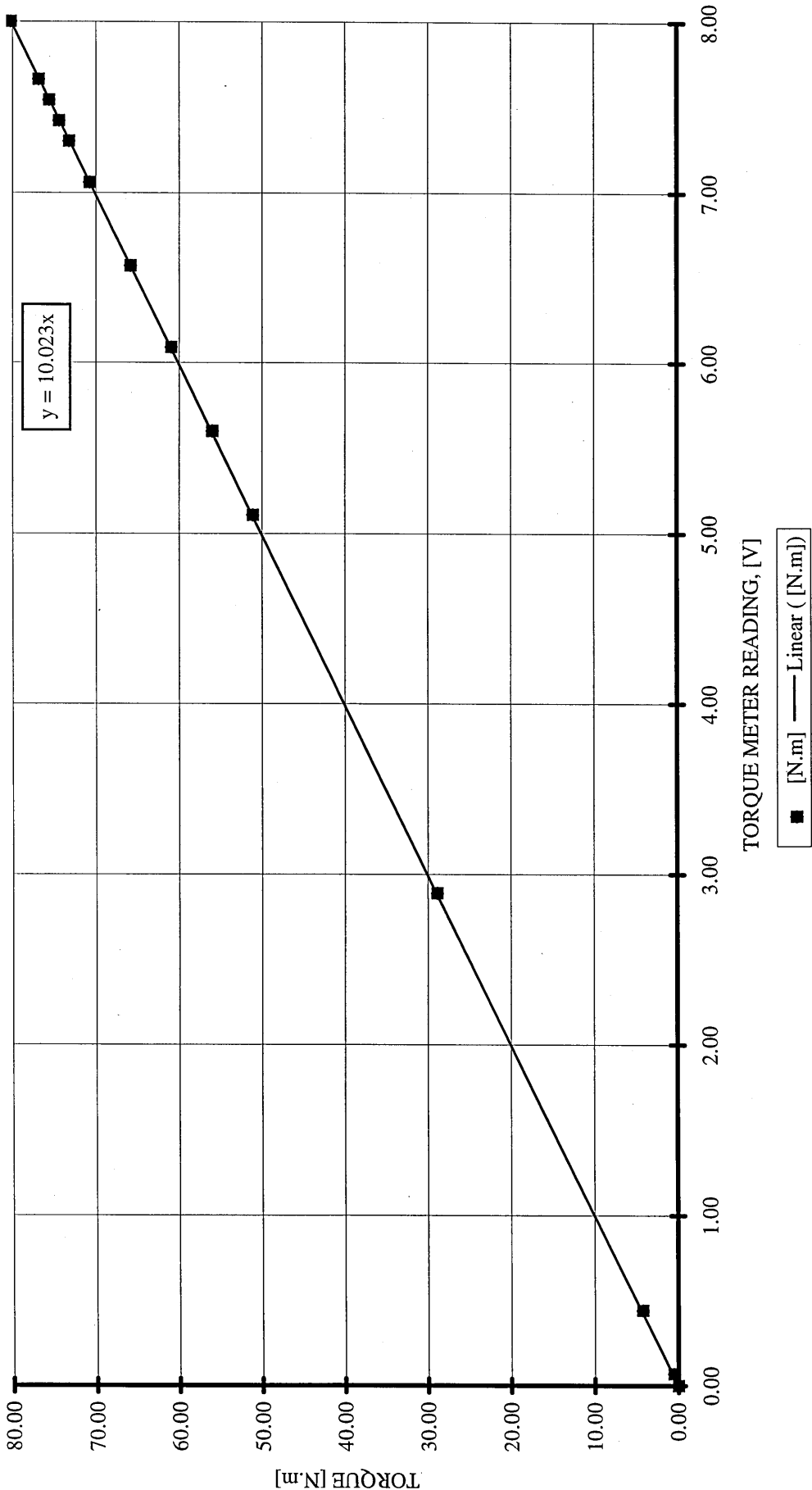
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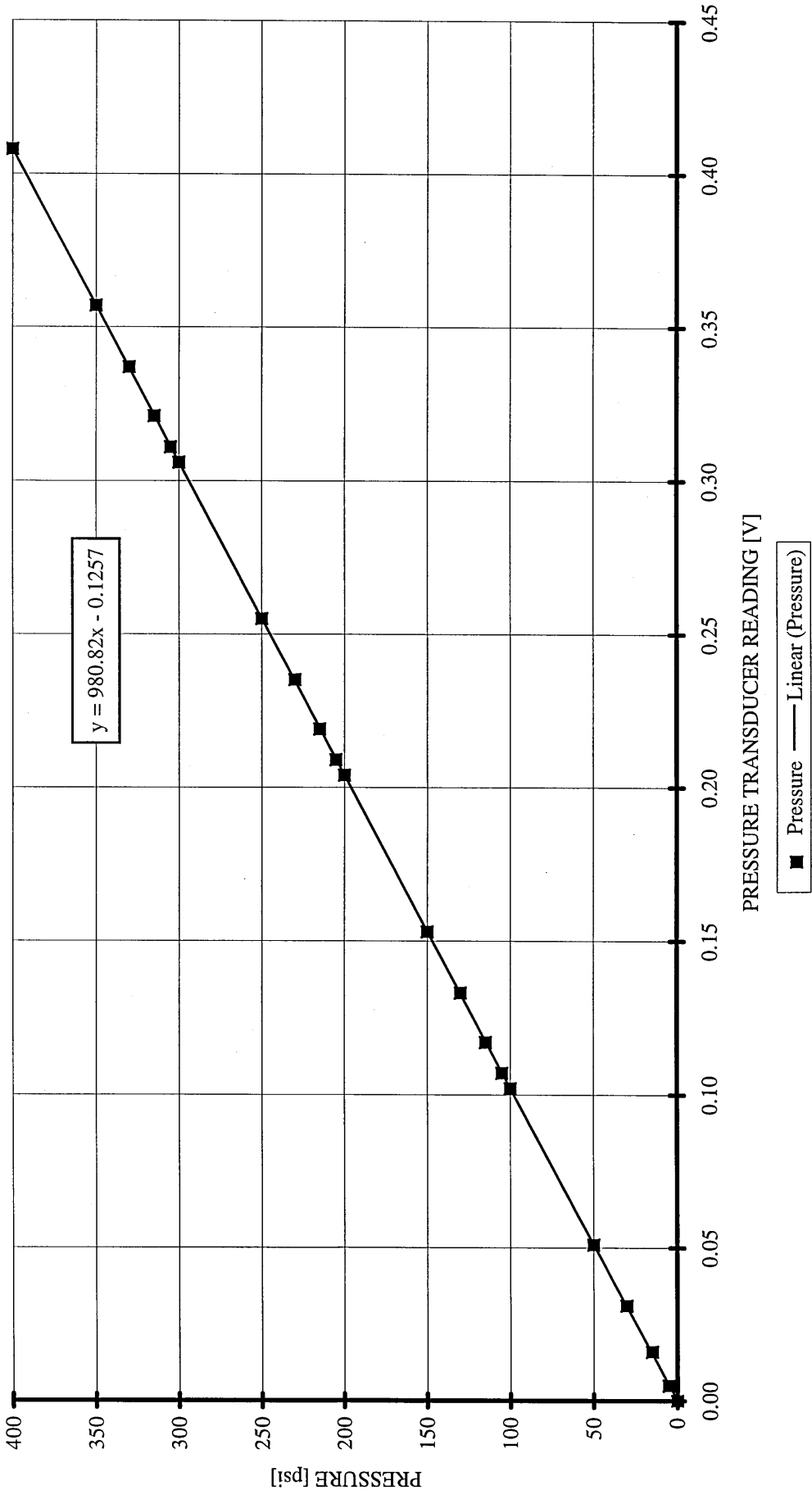
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APPENDICES

APPENDIX 1.1
TORQUE METER CALIBRATION CURVE

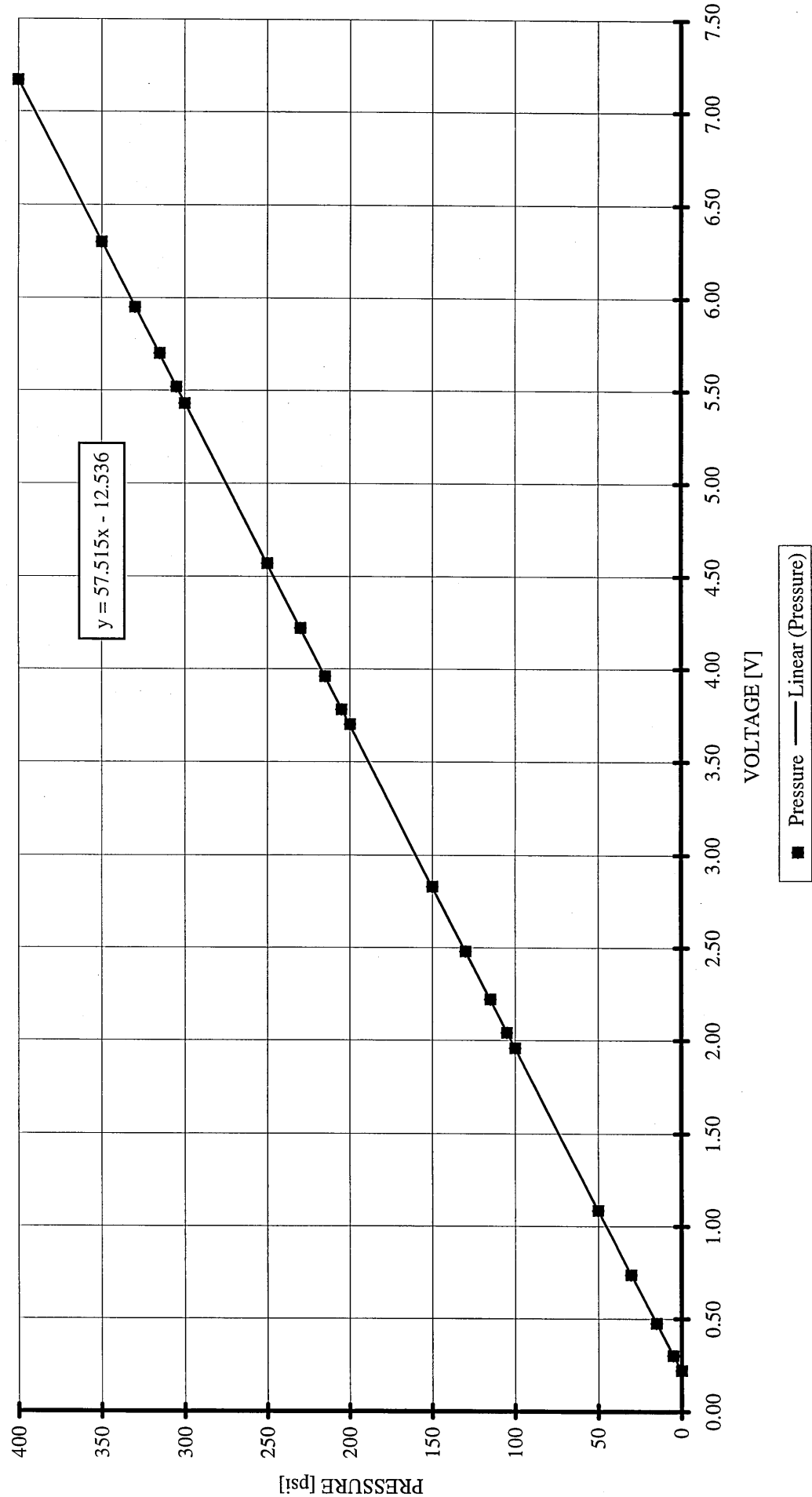


APPENDIX 1.2
PRESSURE TRANSDUCER CALIBRATION CURVE (PRIMARY PULLEY)

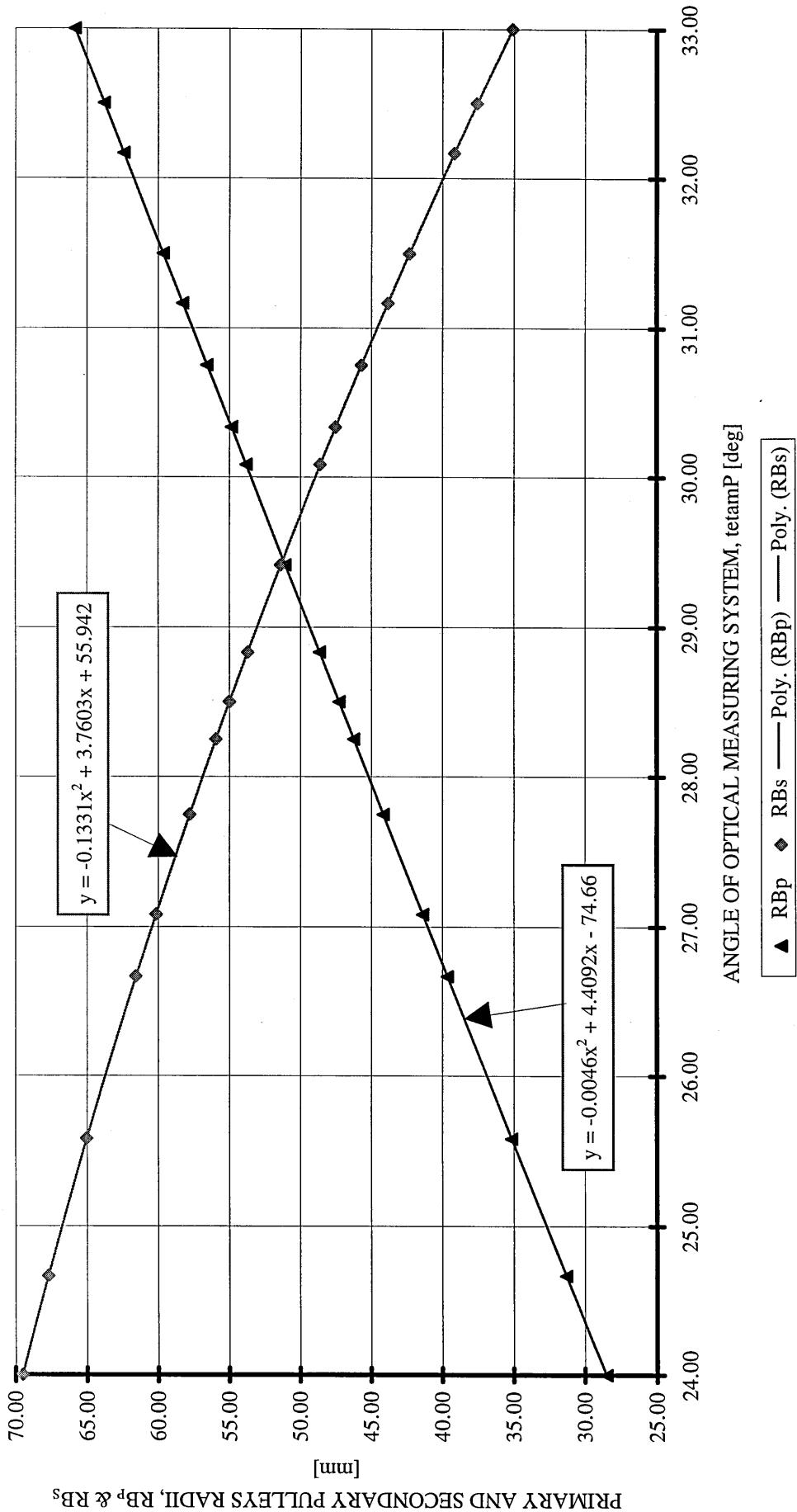


APPENDIX 1.3

PRESSURE TRANSDUCER CALIBRATION CURVE (SECONDARY PULLEY)

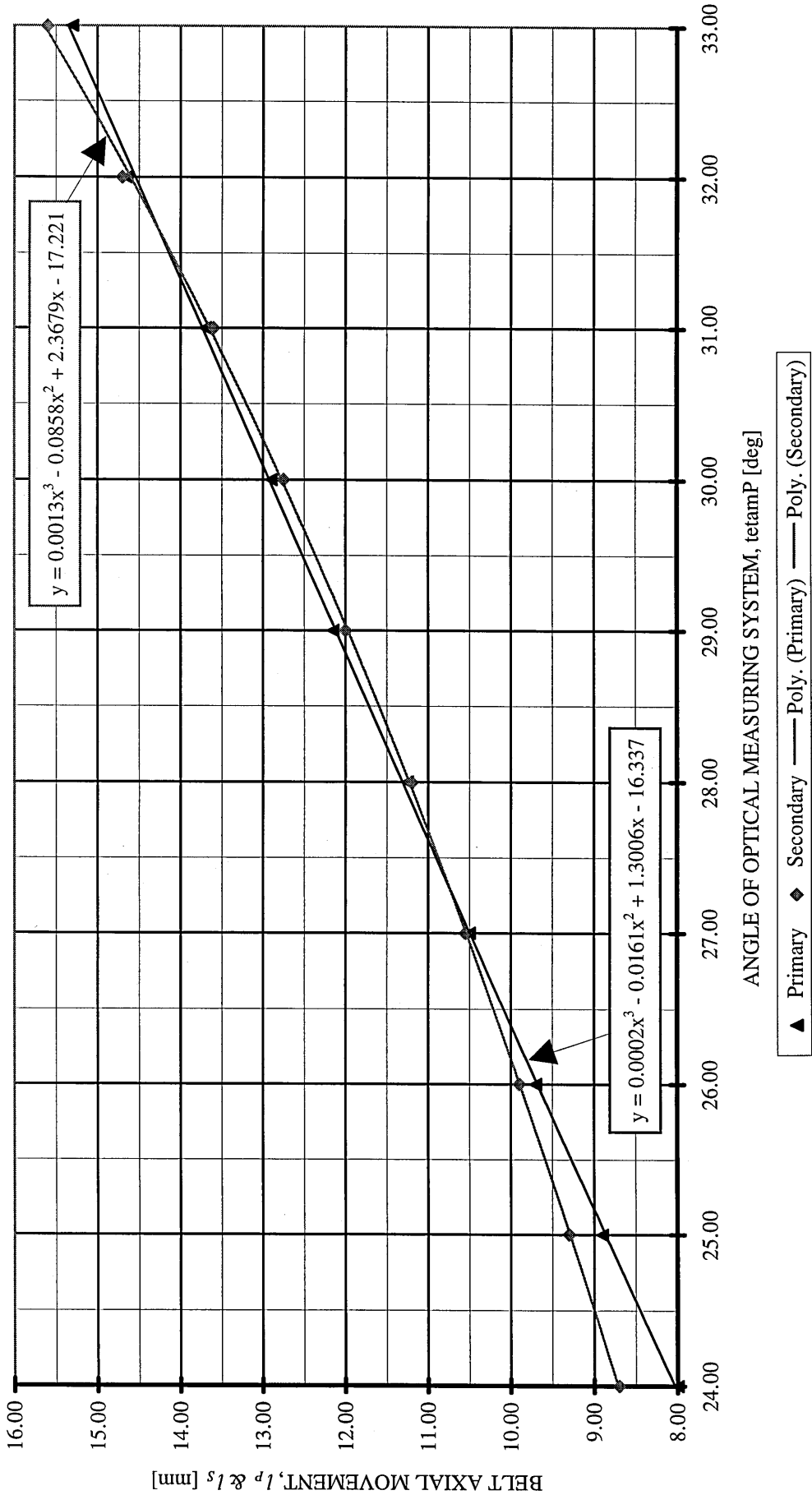


APPENDIX 1.4
BELT RADI CALIBRATION CURVES

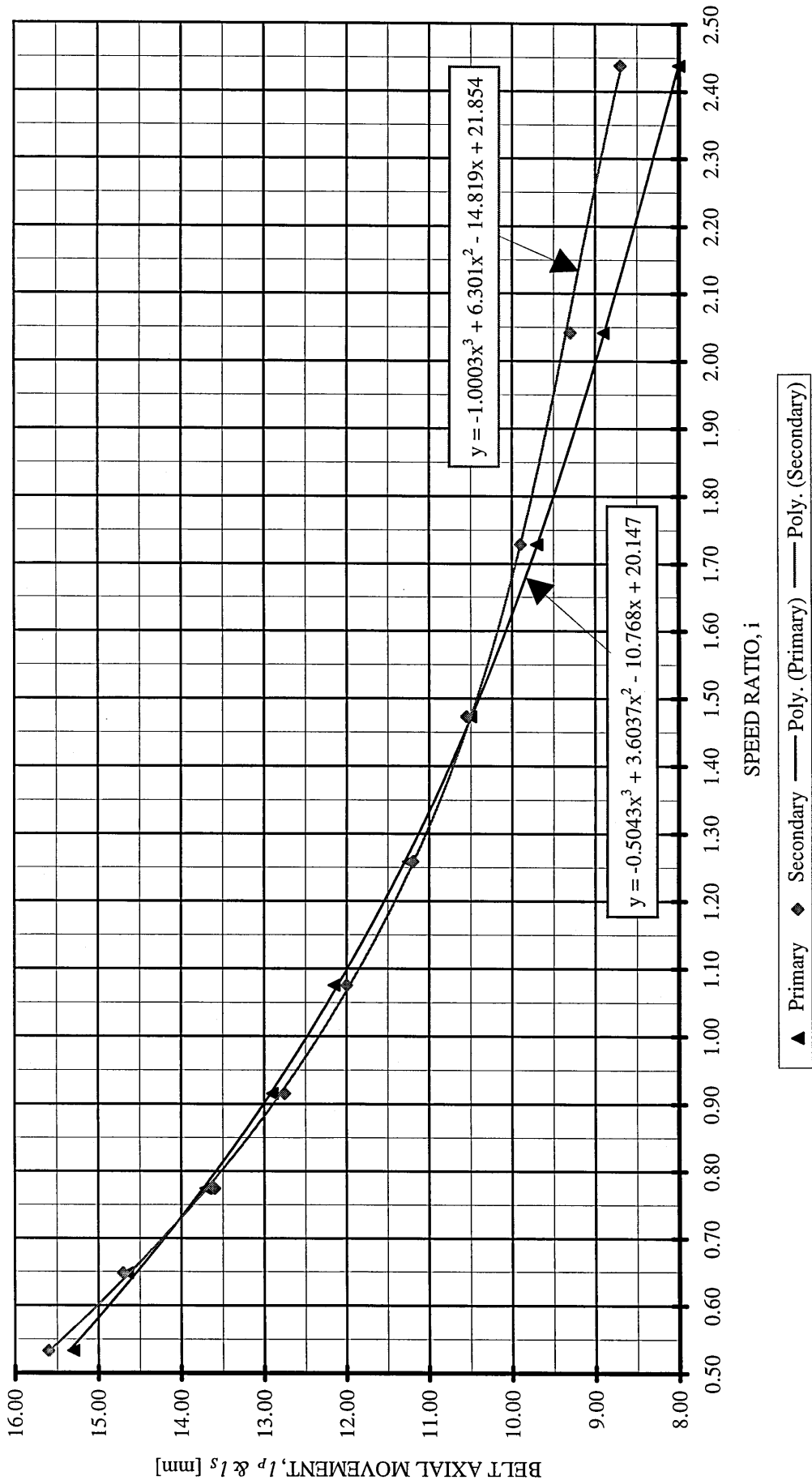


APPENDIX 1.5

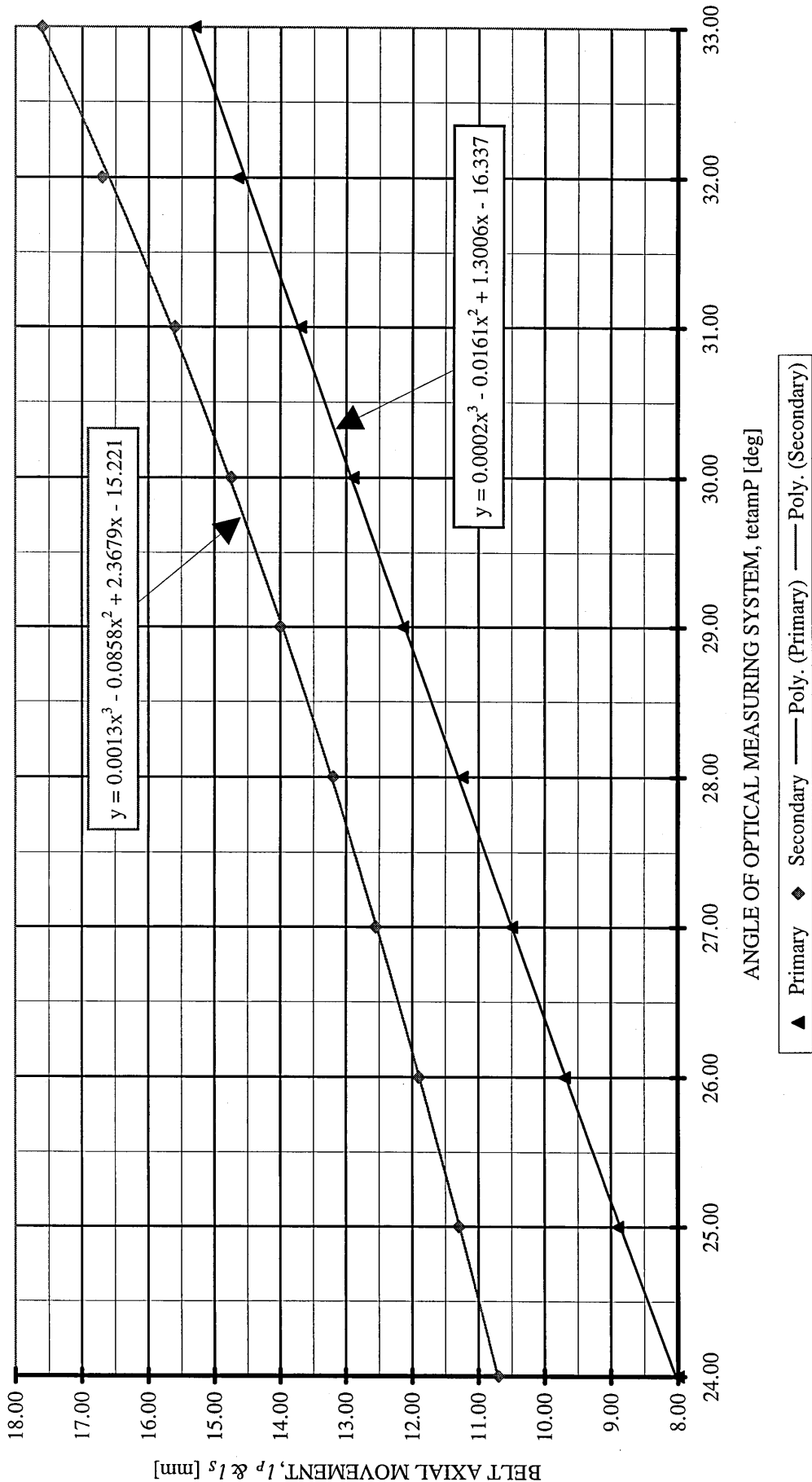
BELT AXIAL MOVEMENT CALIBRATION CURVES (delta = 0.00 mm.)



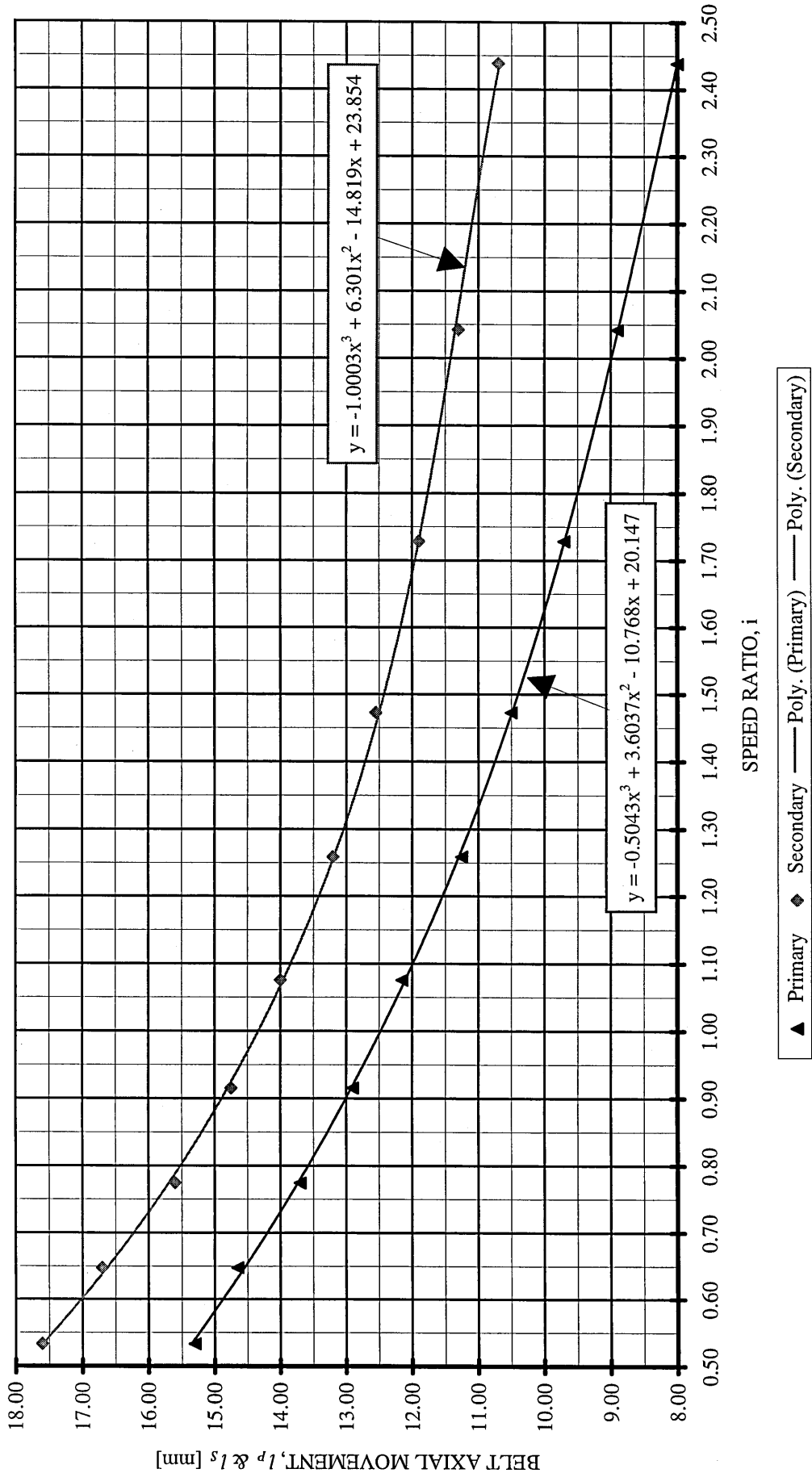
APPENDIX 1.6
BELT AXIAL MOVEMENT CALIBRATION CURVES (delta = 0.00 mm.)



APPENDIX 1.7
BELT AXIAL MOVEMENT CALIBRATION CURVES (adelta = 2.00 mm.)

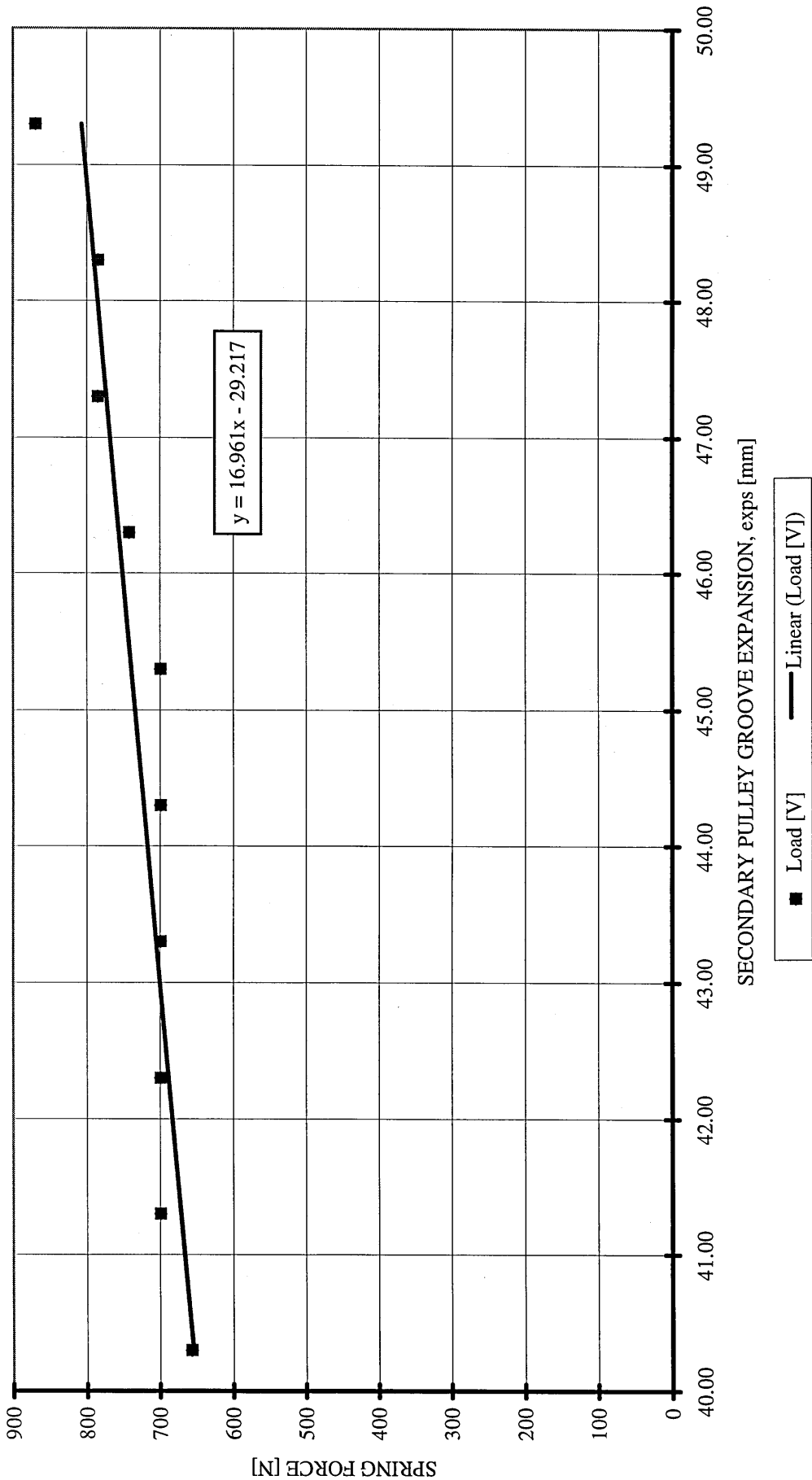


APPENDIX 1.8
BELT AXIAL MOVEMENT CALIBRATION CURVES (delta = 2.00 mm.)

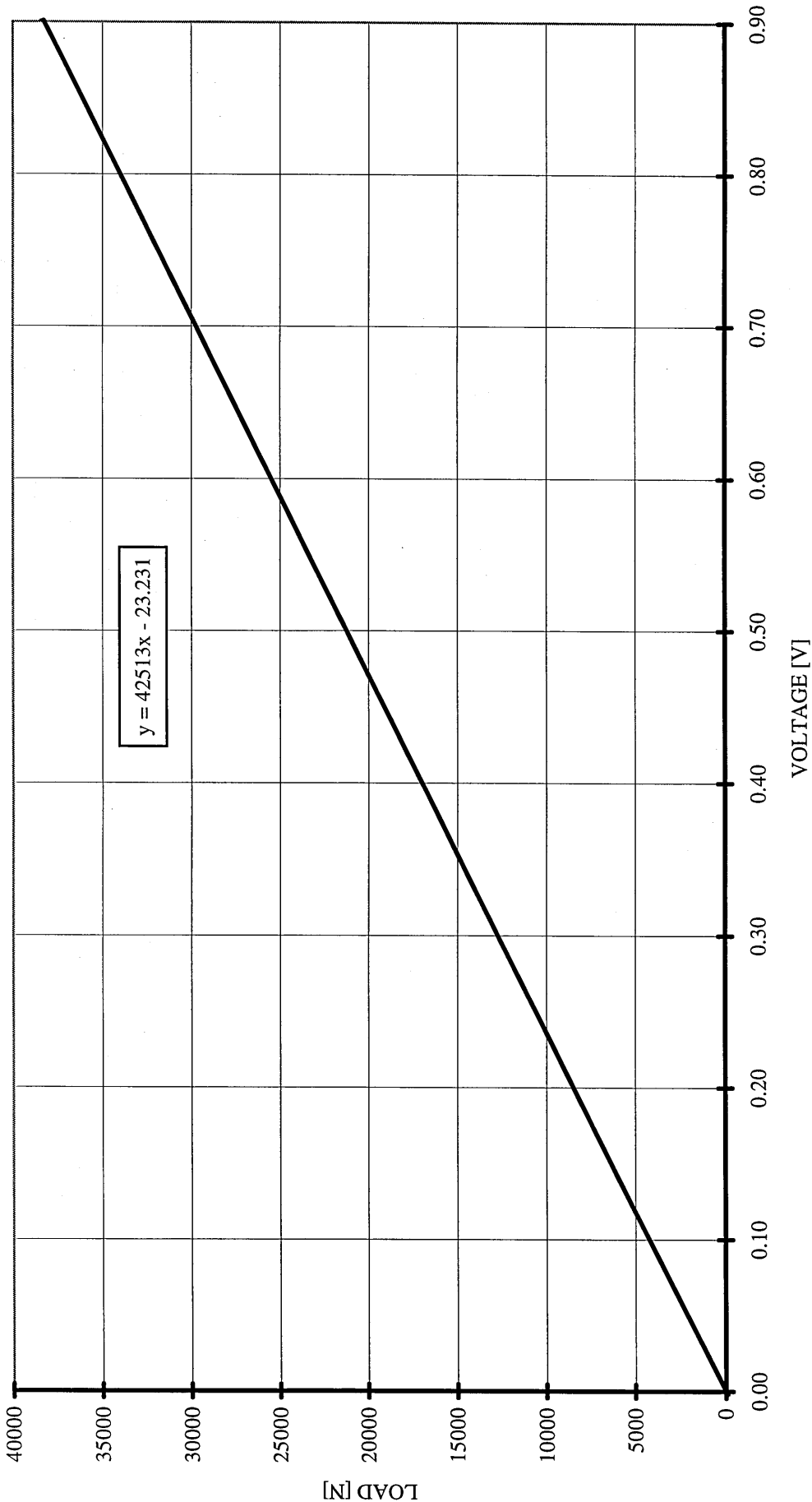


APPENDIX 1.9

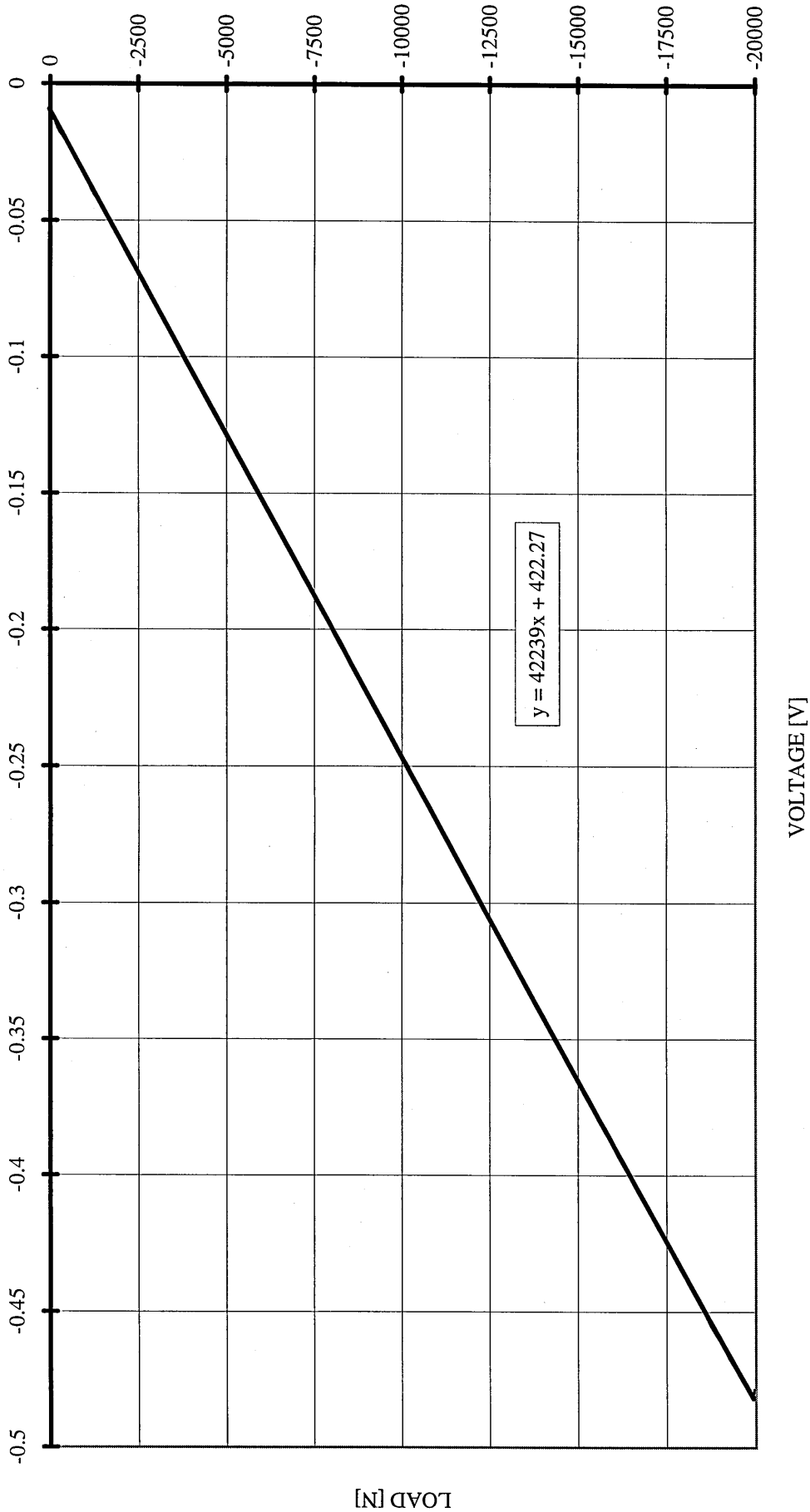
CALIBRATION CURVE FOR LOAD CELL AGAINST SECONDARY PULLEY GROOVE EXPANSION



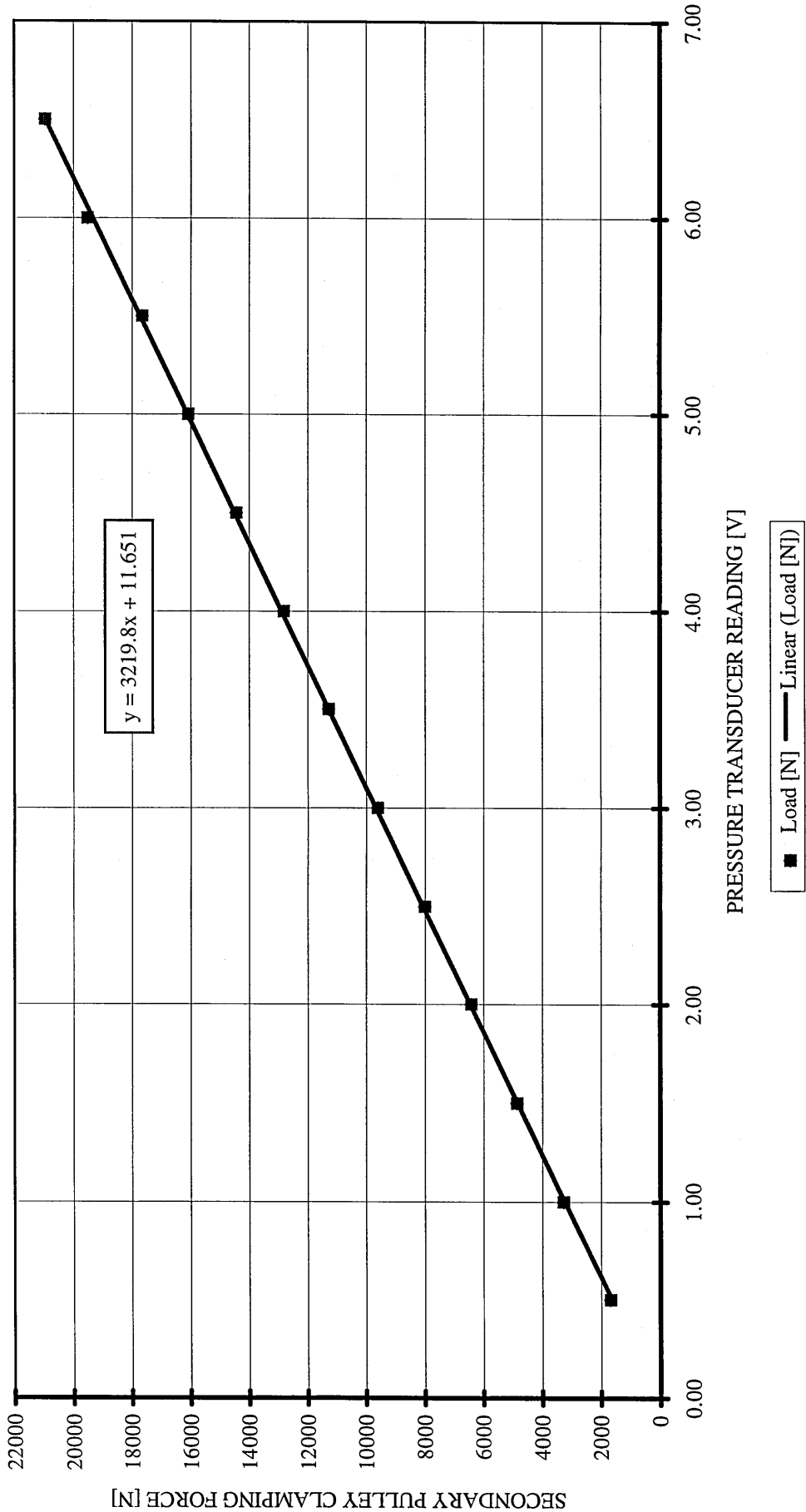
APPENDIX 1.10
LOAD CELL CALIBRATION CURVE (TENSION)



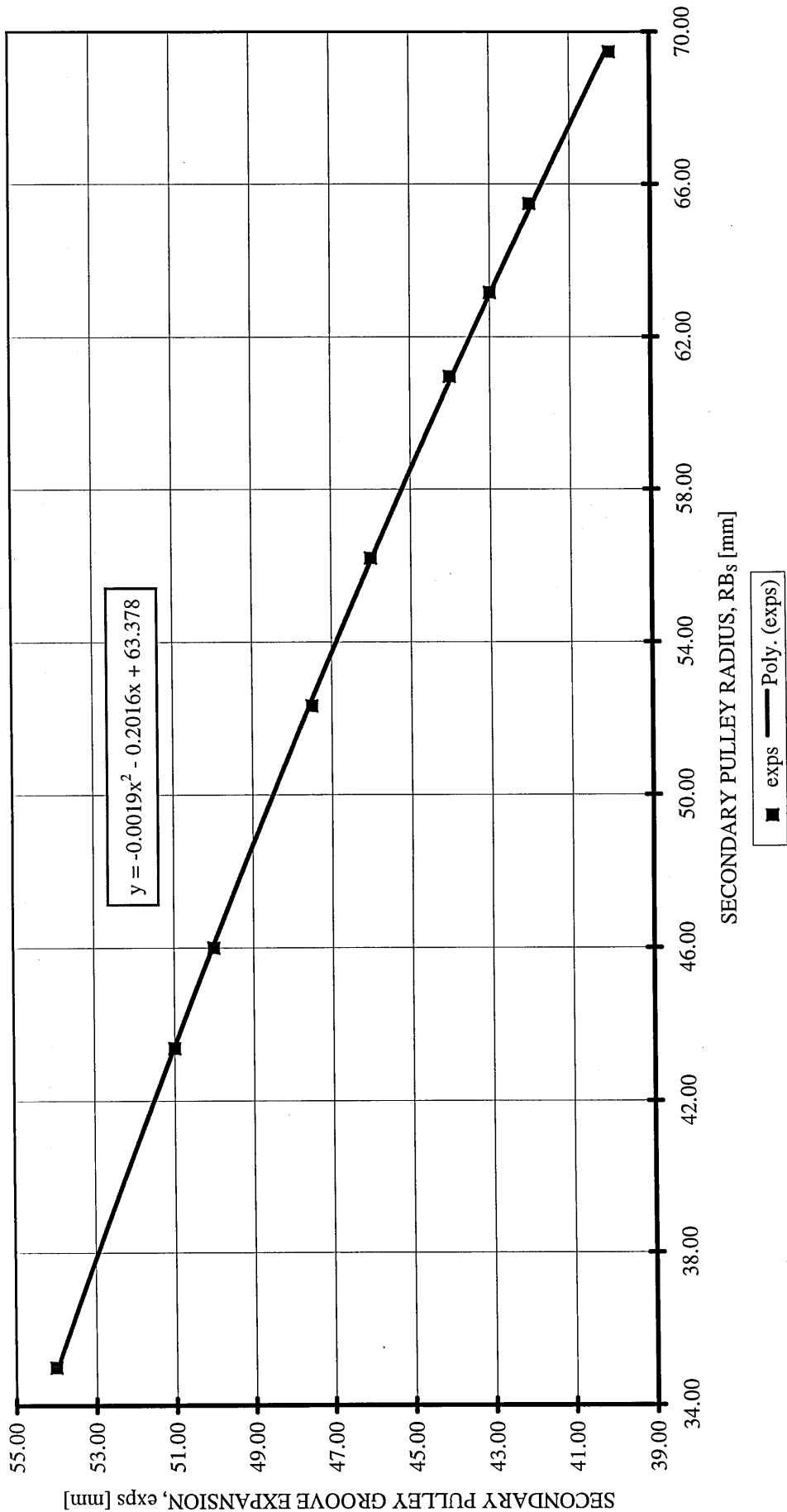
APPENDIX 1.11
LOAD CELL CALIBRATION CURVE (COMPRESSION)



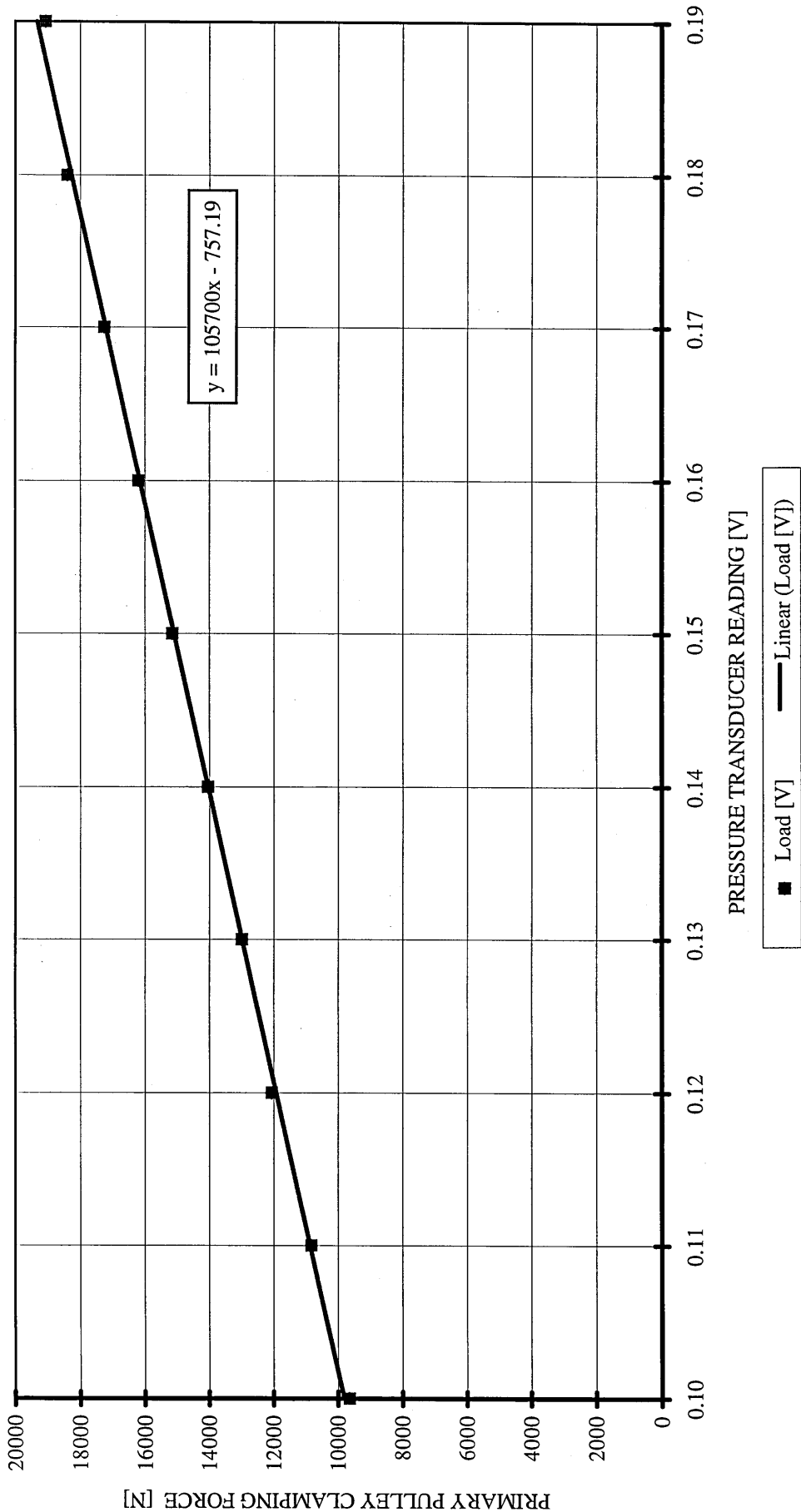
APPENDIX 1.12
CALIBRATION CURVE FOR SECONDARY PULLEY CLAMPING FORCE AGAINST
PRESSURE TRANSDUCER (SECONDARY PULLEY GROOVE EXPANSION = 49.3 mm)



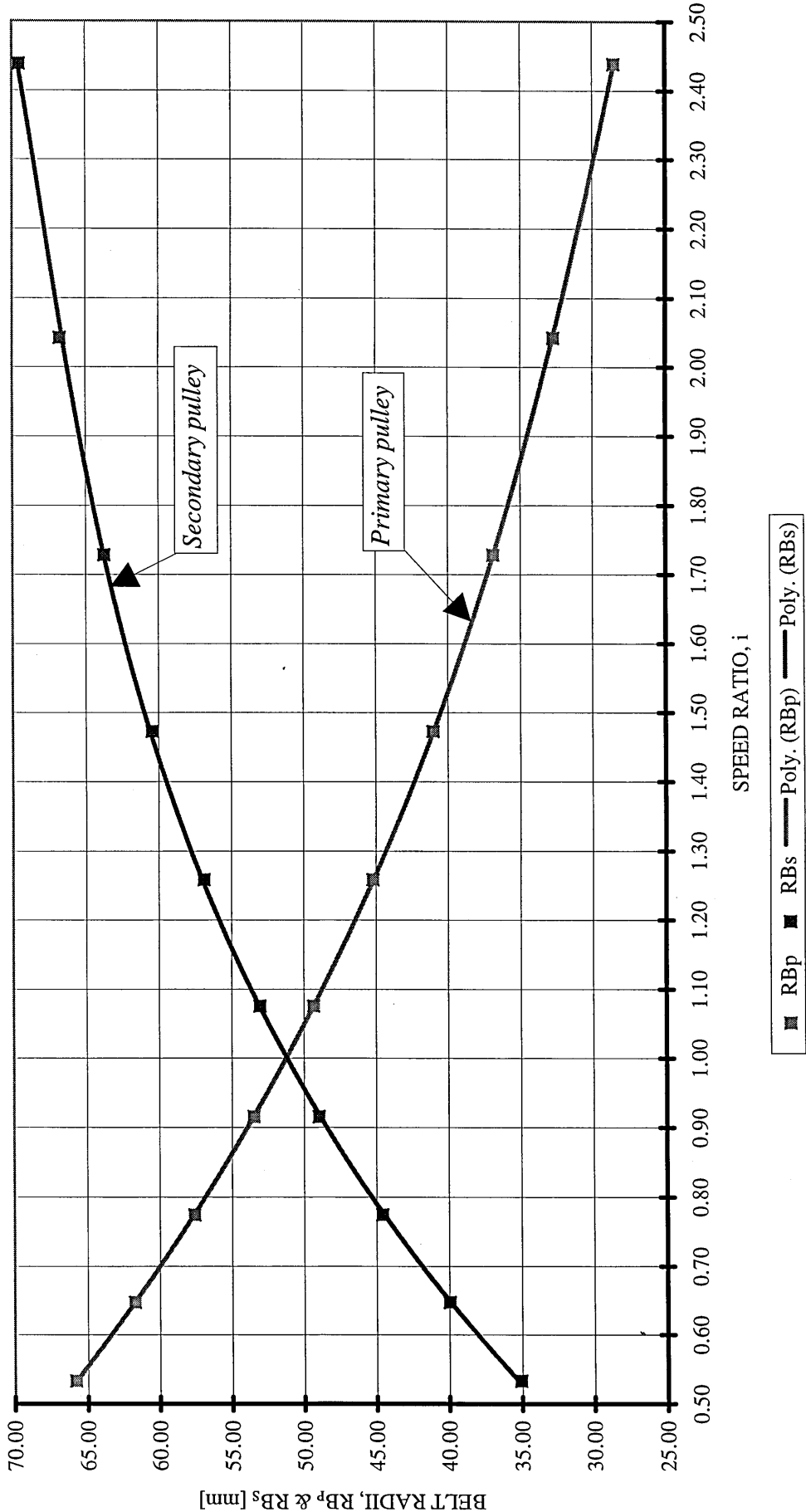
APPENDIX 1.13
CALIBRATION CURVE FOR THE SECONDARY PULLEY GROOVE EXPANSION
AGAINST ITS BELT RADI



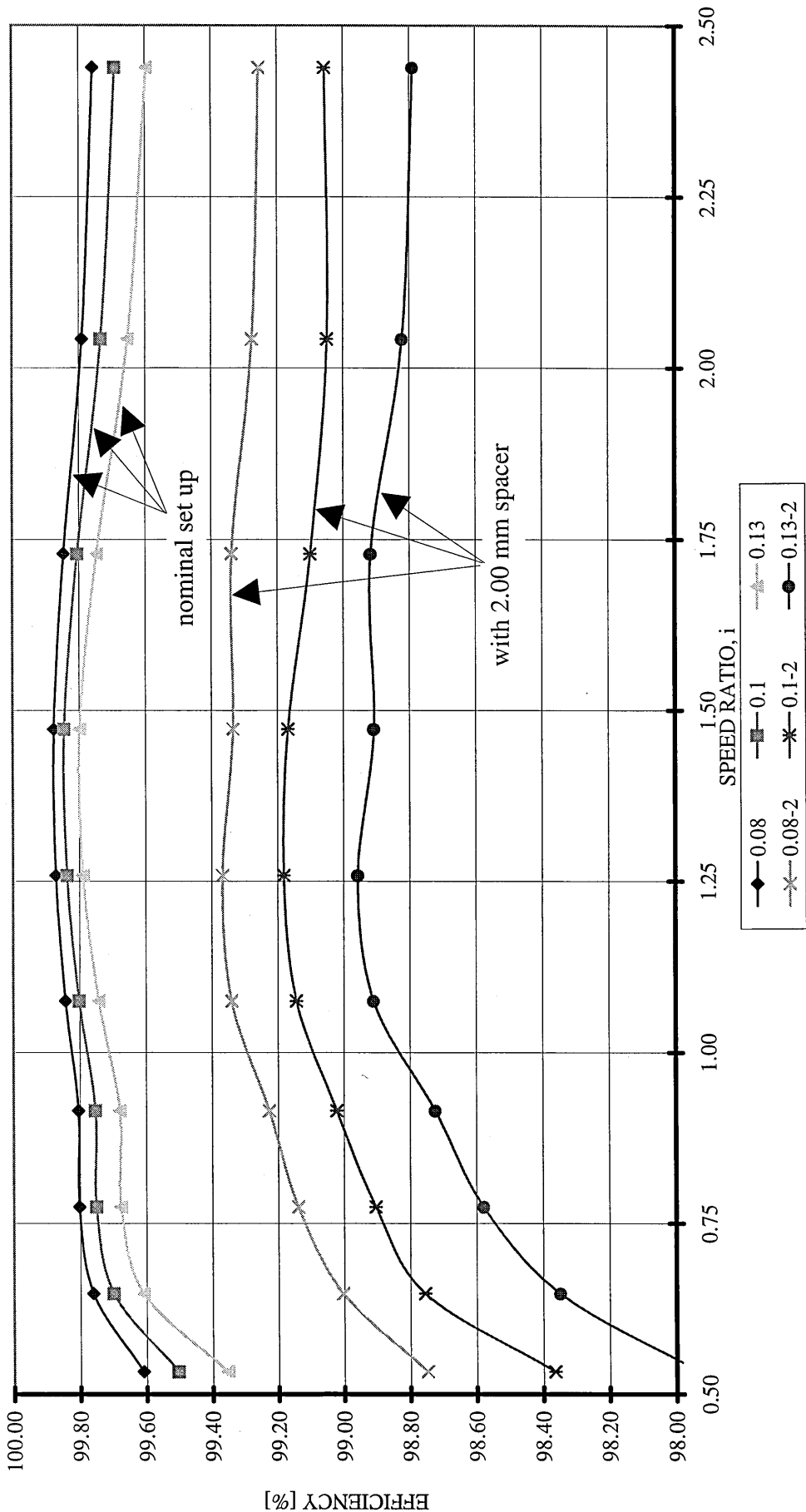
APPENDIX 1.14
CALIBRATION CURVE FOR THE PRIMARY PULLEY CLAMPING FORCE
AGAINST PRESSURE TRANSDUCER



APPENDIX 1.15
BELT RADII GRAPHS AGAINST SPEED RATIO

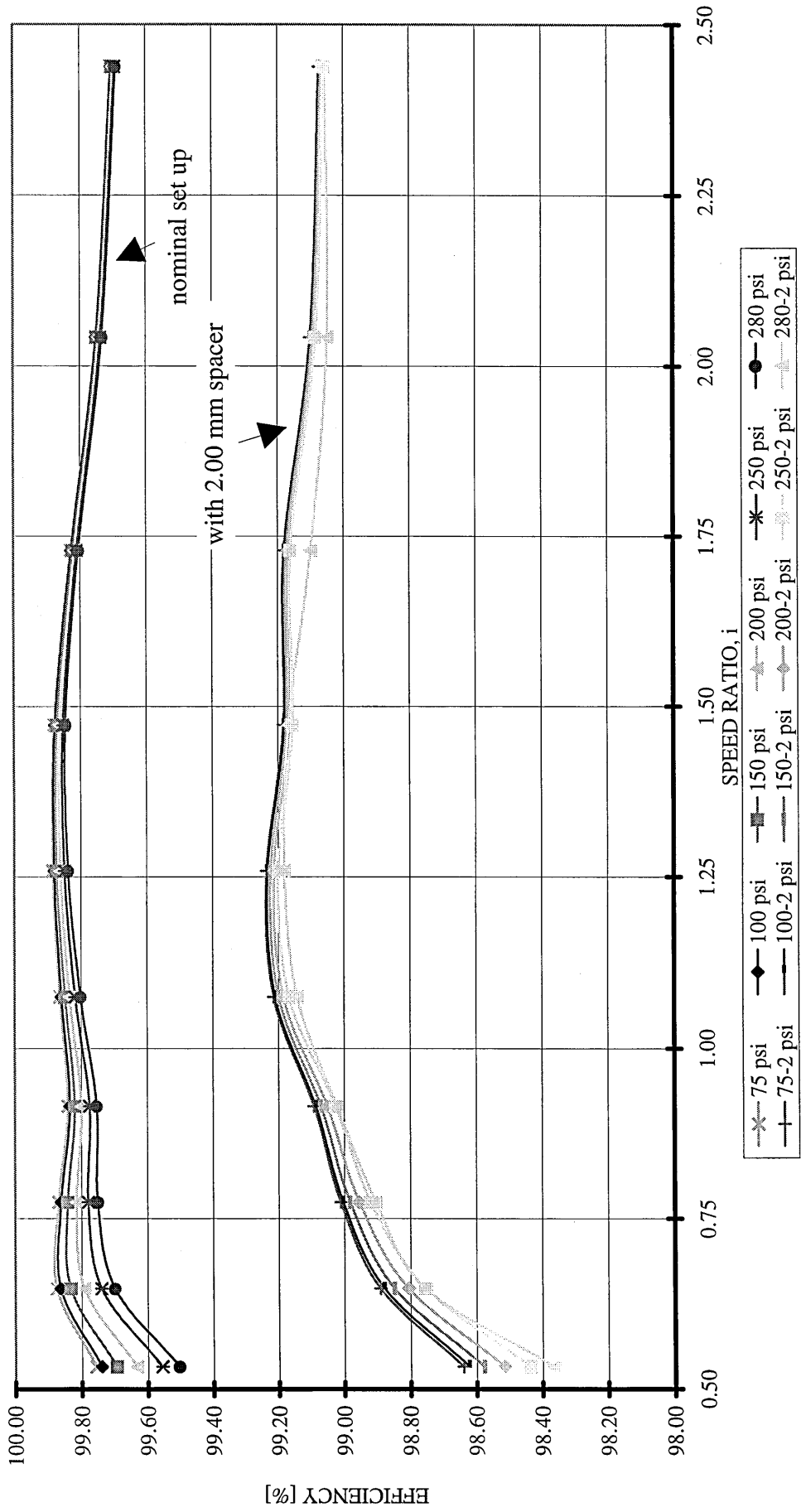


APPENDIX 1.16
Anticipated transmission efficiency (Calculated, Constant input torque = 17 N.m) - Expt. 4a32 and 4b32
(Input speed = 1250 rpm at various coefficient of friction values)



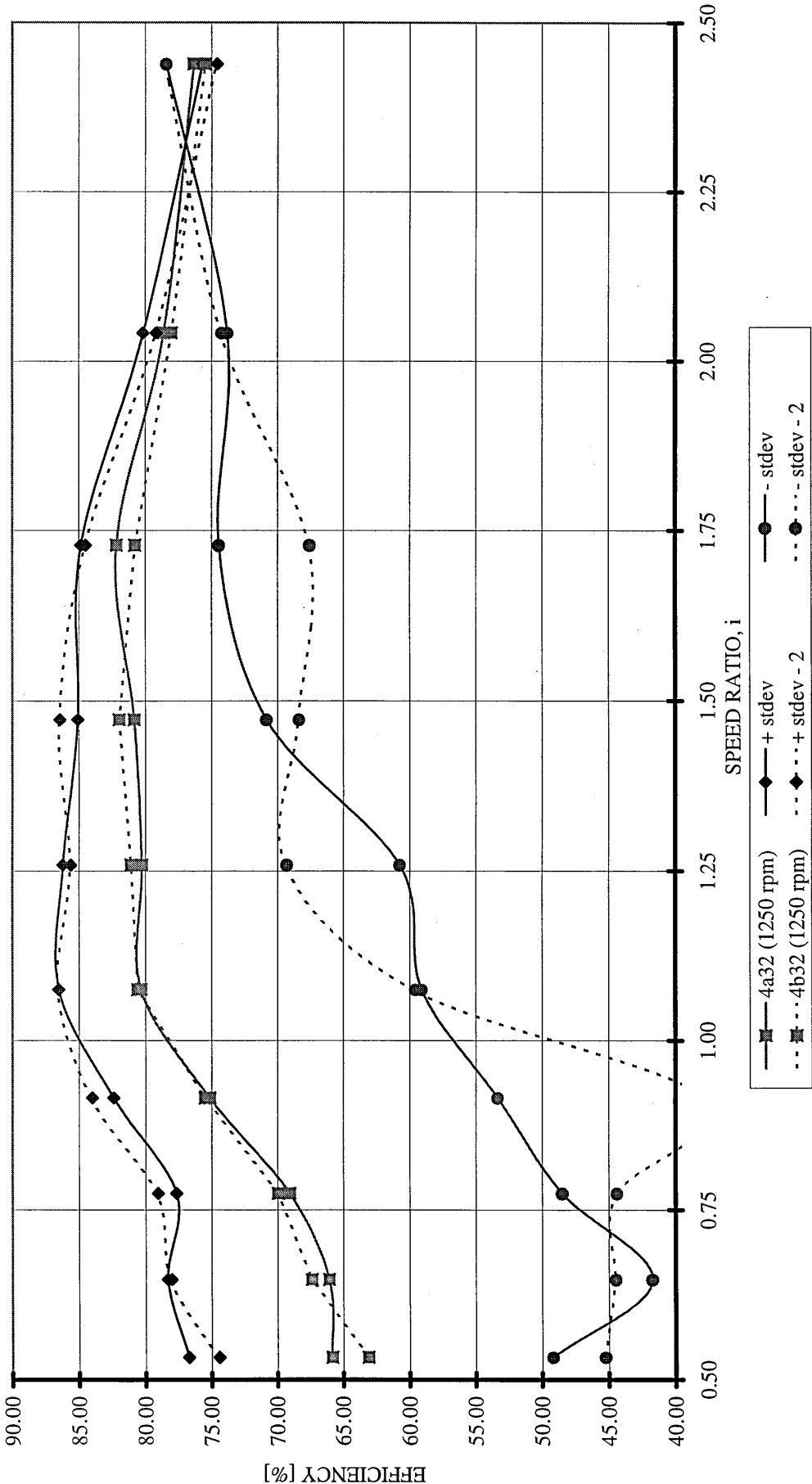
APPENDIX 1.17

Anticipated transmission efficiency (Calculated, Constant input torque = 17 N.m) - Expt. 4a32 and 4b32
(Input speed = 1250 rpm at various secondary pulley pressure)



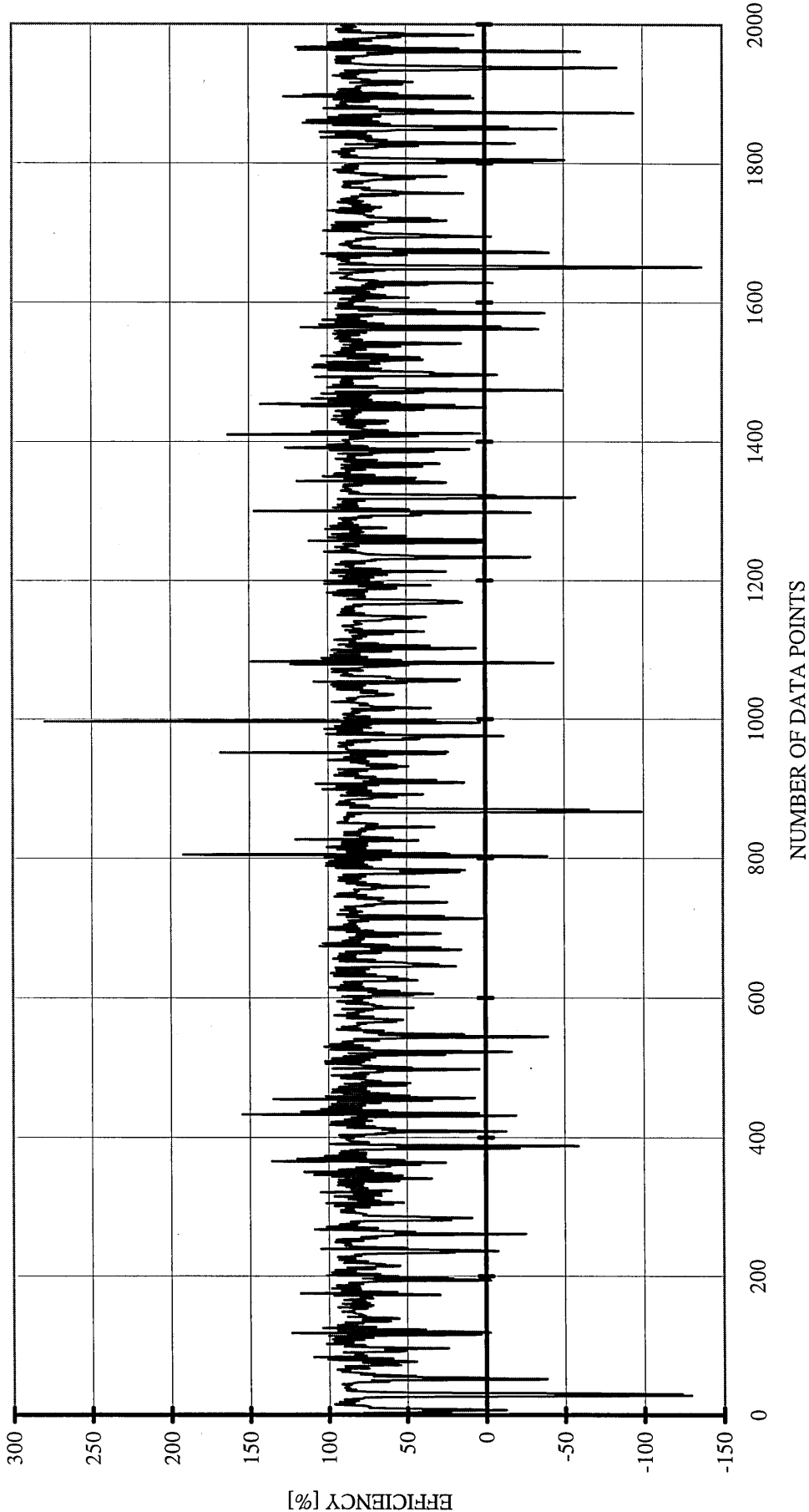
APPENDIX 1.18

Transmission upper and lower efficiency limits - standard deviation
(Measured, Constant input torque = 17 N.m) - Expt. 4a32 and 4b32



APPENDIX 1.19

EFFICIENCY GRAPH FOR Expt. 4a32 AT SPEED RATIO 1.0753 (approximately 81 %)



APPENDIX 1.20

POWER LOSS CALCULATIONS DUE TO BEARINGS LOSS

Introduction

In this appendix, power loss calculations due to the test rig bearing frictional losses and their effect towards the transmission efficiency with regards to belt misalignment, for experiment Expt. 4a32 (nominal set up) at zero (speed ratio of 1.40) and maximum belt misalignments ($\Delta_{max} = 2.7 \text{ mm}$) are shown.

Zero Belt Misalignment

Figure A1 shows the free body diagrams of the transmission for both primary and secondary pulley shafts. The values of the belt tensions were taken from App. 1.21.

Primary Pulley Shaft

Resolving forces along the y-axis of the primary pulley, we get,

$$\sum F_y: A_{Py} + B_{Py} = 3185$$

Taking moments about reaction A_{Py} , we get,

$$\sum M_A: 125 \cdot B_{Py} = 3185 \cdot (90)$$

$$B_{Py} = \frac{3185 \cdot (90)}{125} = 2293.2 \text{ N}$$

$$\therefore A_{Py} = 891.8 \text{ N}$$

From SKF general catalogue for the deep groove ball bearing at point B_P , the estimation of frictional moment is given by,

$$M_{PB} = 0.5 \cdot \mu \cdot F \cdot d \text{ [N.mm]}$$

where, $\mu = 0.0015$, $F = B_{Py}$ and $d = 30 \text{ mm}$ therefore,

$$M_{PB} = 0.5 \cdot (0.0015) \cdot (2293.2) \cdot (30) = 51.60 \text{ [N.mm]}$$

For the deep groove ball bearing at point A_P , with $F = A_{Py}$ and $d = 30 \text{ mm}$, the estimation of frictional moment is equal to,

$$M_{PA} = 0.5 \cdot (0.0015) \cdot (891.8) \cdot (30) = 20.07 \text{ [N.mm]}$$

Bearing power loss PL_{BrgP} , at $\omega_p = 1250 \text{ rpm}$ at the primary pulley is equal to,

$$\begin{aligned} PL_{BrgP} &= (M_{PA} + M_{PB}) \cdot \omega_p \\ &= \left(\frac{51.60 + 20.07}{1000} \right) \cdot \left[\frac{(1250) \cdot (2) \cdot \pi}{60} \right] \\ &= 9.38 \text{ W} \end{aligned}$$

Secondary Pulley Shaft

Resolving forces along the y-axis, we get,

$$\sum F_y: A_{Sy} + B_{Sy} = 3185$$

Taking moments about reaction A_{Sy} , we get,

$$\sum M_A: 196 \cdot B_{Sy} = 3185 \cdot (36)$$

$$B_{Sy} = \frac{3185 \cdot (36)}{196} = 585 \text{ N}$$

$$\therefore A_{Sy} = 2600 \text{ N}$$

For the deep groove ball bearing at point B_S , with $F = B_{Sy}$ and $d = 40 \text{ mm}$, the estimation of frictional moment is equal to,

$$M_{SB} = 0.5 \cdot (0.0015) \cdot (585) \cdot (40) = 17.55 \text{ [N.mm]}$$

For the deep groove ball bearing at point A_S , with $F = A_{Sy}$ and $d = 25 \text{ mm}$, the estimation of frictional moment is equal to,

$$M_{SA} = 0.5 \cdot (0.0015) \cdot (2600) \cdot (25) = 48.75 \text{ [N.mm]}$$

Bearing power loss PL_{BrgS} , at $\omega_s = \frac{1250}{i} = \frac{1250}{1.40} \approx 893 \text{ rpm}$ at the secondary pulley is equal to,

$$\begin{aligned}
 PL_{BrGS} &= (M_{SA} + M_{SB}) \cdot \omega_s \\
 &= \left(\frac{48.75 + 17.55}{1000} \right) \cdot \left[\frac{1250}{1.40} \cdot \frac{(2) \cdot \pi}{60} \right] \\
 &= 6.20 \text{ W}
 \end{aligned}$$

Hence, the total bearing loss for the transmission PL_{BrG} , is equal to

$$\begin{aligned}
 PL_{BrG} &= PL_{BrGP} + PL_{BrGS} \\
 &= 9.38 + 6.20 \\
 &= 15.58 \text{ W}
 \end{aligned}$$

Therefore the measured transmission efficiency η , will increase to,

$$\eta = \left(1 - \frac{TPL_{meas.} - PL_{BrG}}{H_P} \right) \cdot 100$$

where, at speed ratio of 1.40 (zero belt misalignment) from App. 1.22, the total measured transmission power loss, $TPL_{meas.} = 412 \text{ W}$ and its input power, $H_P = 2130 \text{ W}$.

Hence,

$$\begin{aligned}
 \eta &= \left(1 - \frac{412 - 15.58}{2130} \right) \cdot 100 \\
 &= 81.39 \%
 \end{aligned}$$

That is, the transmission efficiency will increase by less than 4 %.

Maximum Belt Misalignment

From Fig. A2, it can clearly be seen that the angle made by the belt due to maximum belt misalignment from the vertical is quite small (about 1.105° only). Hence the reduction in belt tension component in the vertical direction is also very small (about 0.02 % only). This in turn resulted to only a small axial force that is about 1.93 % of the belt radial force. Therefore the change in bearing loads was negligible and the change in friction loss in the bearings is also negligible.

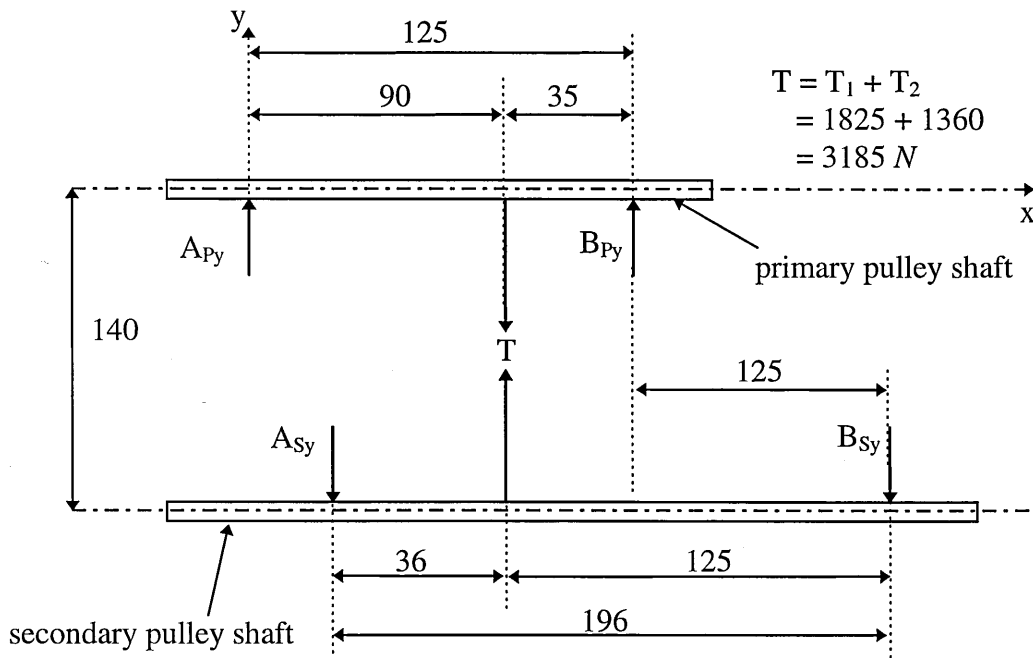


Fig. A1 Free body diagram of primary and secondary pulley shafts at zero belt misalignment at speed ratio 1.40. Dimension in mm

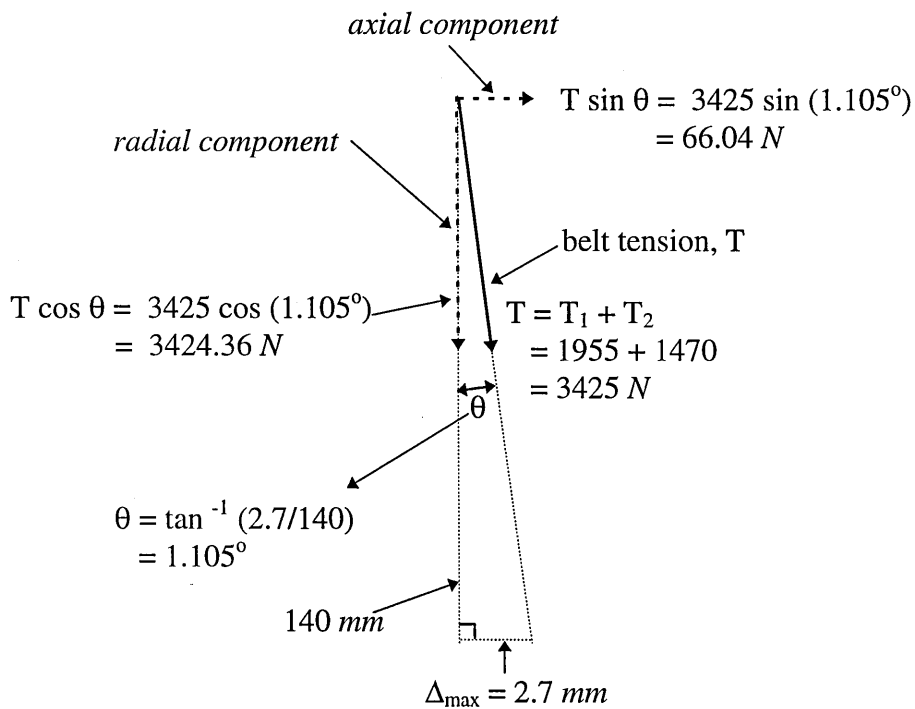
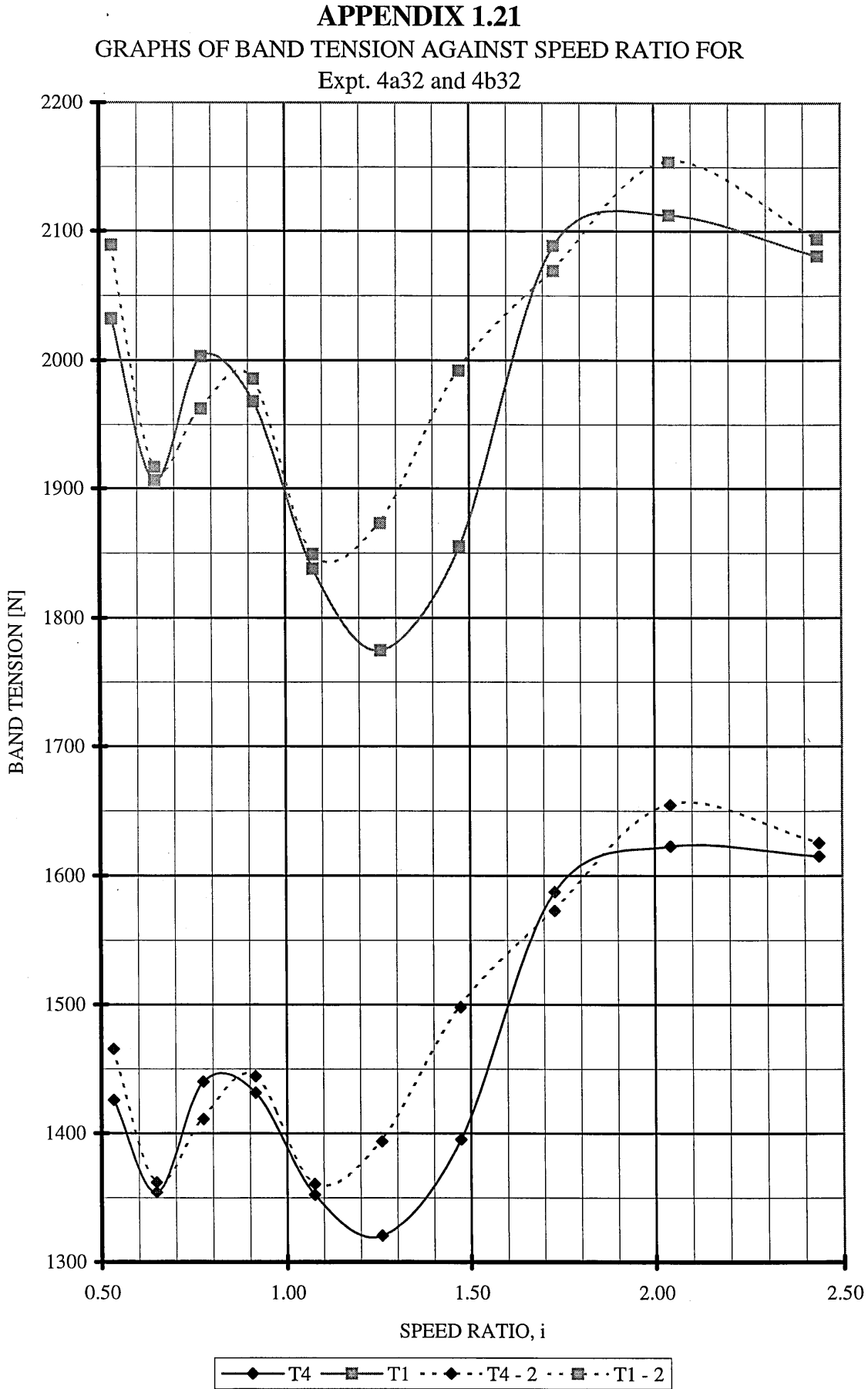


Fig. A2 Free body diagram of belt tension at maximum belt misalignment



APPENDIX 1.22

GRAPHS OF MEASURED TRANSMISSION POWER LOSS AND INPUT
POWER AGAINST SPEED RATIO FOR Expt. 4a32

