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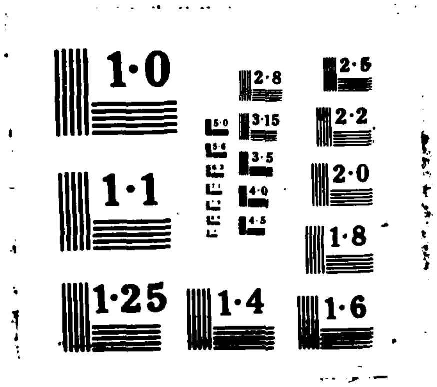
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Vibration Control in Rotating Machinery Using Variable Dynamic Stiffness Squeeze-Films

1st Full Interim Scientific Report covering the period 1Sept 1984-31 August 1985 VOL I

M P Roach
M J Goodwin

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This report describes the current status of a research project whose aim is to develop a hydrostatic bearing, for rotating machinery, whose dynamic characteristics may be tuned during operation of the machine. The purpose of this is to enable the operator to exercise some control over machine critical speeds and vibrations.

A computer program has been written which will predict both the static and dynamic characteristics of a hydrostatic bearing. The program allows for the presence of accumulators linked to the hydrostatic bearing recesses via flow restrictors. Output from the computer program has been used as input data to a second computer program which calculates machine vibration amplitude variation with running speed. Theoretical machine characteristics obtained in this way have been used to aid the design of a test rig which will be used to examine the practical performance of the new bearing type being developed.

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CONTENTS

LITERATURE SURVEY

LITERATURE SURVEY

THE ORIGINS OF LUBRICATION THEORY

After Towers' (1) experimental findings concerning the existence of pressures significantly above atmospheric pressure in the oil film of a journal bearing, Reynolds (2) developed equations to predict the pressure distribution due to the viscose forces. This was the first theoretical explanation of a practical phenomena which had been unknowingly used for years.

To ease the problem of boundary conditions required for the solution of Reynolds equations, Ocvirk (3) produced a short bearing approximation with a 180 degree positive pressure arc and circumferential flow neglected and Sommerfeld produced a long bearing approximation by assuming negligible pressure variation in the axial direction. These works formed the foundation for later developments in lubrication theory.

EARLY DEVELOPMENTS IN ROTOR DYNAMICS

In 1869 Rankine (4) mathematically examined the equilibrium conditions of a mass mounted on a flexible rotating shaft supported in frictionless bearings. Due to his neglect of Coriolis' forces, he concluded that the rotor is always unstable above the shaft critical speed. Later in 1885 de Laval experimentally demonstrated that super critical operation was possible but manufacturers experienced many design failures due to excessive vibration at supercritical speeds. Dunkerley (7) in 1894 using a multi-mass, multi-stiffness model of a shaft system, theoretically predicted the ranges of "critical speeds" characterised by rough running and large force transmission to the foundations.

Jeffcott (5) justified the use of an elegantly simple single mass, flexible rotor model by revealing the mechanism of shaft resonance. He concluded that below the shaft pin-pin critical speed the rotor centre precesses at the same angular velocity and in phase with the shaft imbalance. This was termed "syncronous precession". At the shaft critical speed this phase difference was shown to be 90° and well above the critical speed the unbalance was 180° out of phase with the shaft centre. The shaft was shown to reduce its flexure amplitude above critical speed due to this "inversion" process (shaft centre line rotating about its mass centre) which led in the 1920's to the use of supercritical rotors, termed flexible.

However, the problem of low frequency shaft whipping whilst operating at supercritical speeds remained a problem and so in 1924 Newkirk (6) was commissioned to experimentally investigate the problem. Using both built-up and non-built-up rotors supported on rolling element bearings he noted that the onset speed of whirling and the whirl amplitude were unaffected by rotor unbalance and always exhibited the same precession speed. The threshold speed could be increased by increasing the foundation flexibility, introducing foundation damping, misaligning the bearings and by applying axial thrust to the bearings. The whirling was found only to occur with built-up rotors. By introducing internal friction into Jeffcotts' model, Newkirk showed that the attenuating effect of internal friction became an exciting force well above critical speed.

Poritsky (8) investigated the phenomena using a similar flexible rotor model but contrary to Newkirk's findings he concluded that the introduction of system foundation flexibility lowers the critical speed of the system and hence reduces the stability threshold.

The importance of foundation flexibility and damping (impedance) on the systems dynamic behaviour had by this time been realised but the theoretical and experimental contradictions as to the effect of changing the support impedance or the reasons for such effects were unresolved. Work by Stodola (10) in 1925 showed that the presence of the oil film journal bearings in an otherwise rigid system would reduce the system's critical speed to well below that calculated for the rigidly supported system. As a consequence of this many authors (11, 12, 14-18) have attempted to accurately model the oil film effect on system response by developing stiffness and damping coefficients to represent the oil film in the journal bearing.

Gunter (13) in 1966 further developed Jeffcotts' model of a single point mass flexible rotor with bearing radial stiffness and internal friction to show the effect of foundation flexibility effects. The interaction of support stiffness assymetry, rotor stiffness and damping factors is considered. From this analysis he showed that for a given value of external damping there is a value of support flexibility which will make the system entirely stable for all speeds. In the absence of damping, assymetric support stiffness is shown to improve stability above that of the symmetric support. However this results in a non-circular procession loci of the shaft centre in the bearing plane with the associated increase in transmitted force at the ellipse tails. The symmetric support produces circular loci (in the stable syncronous zone) and with the addition of correct support damping, increases the stability threshold to its highest value.

The assumption of negligible gyroscopic forces is generally accepted but the single mass model does limit the models representation of practical rotor systems. No new experimental results are provided but Gunter does

compare his theoretical results with the experimental work of Newkirk and of Pinkus (7) and explains some of the apparent anomalies of their findings.

DOUBLE BEARINGS

Some on the first work on "double bearings" was carried out by Shaw and Nussdorfer in 1947 (19). They developed a form of double journal bearing with a floating ring in between the two oil films. The object of the bearing was to reduce the high losses associated with the usual single film bearing and to increase oil flow to allow cooler running conditions. The shaft was centralised by a cantilever mechanism and the floating centre ring was free to rotate. Unfortunately, despite achieving their design objectives, they encountered severe instability of the bearing and startup ring seizure when the bearing was statically loaded, a problem which remained unsolved.

In 1968 Orcutt and Ng (20) investigated a double (floating ring) bearing similar to that of Shaw and Nussdorfer. In particular they investigated the effect of supply pressure, relative film clearance ratio and lubricant viscosity on bearing performance. Whirl instability within the design operating range again limited the number of results taken. However stability was improved by bearing oil flow restriction, viscosity increase, avoidance of excessively high supply pressures and by lowering the inner to outer film clearance ratio from 1.3 to 0.7 (shown by comparing whirl amplitudes).

The stability improvements were not however of sufficient magnitude to completely solve the instability problems encountered.

SQUEEZE FILM DEVELOPMENT

To vary the support characteristics of rotor systems, squeeze films can be introduced around the bearing system used. The stiffness and damping properties of the oil film are used to attenuate the vibrational amplitudes and the forces transmitted to the foundations, and to increase the stability threshold speed of the system. However these factors are sometimes conflicting.

There are two main design techniques used. First, the spring centralised squeeze film which uses a squirrel cage type spring or similar to overcome the shaft gravity load and hence provide zero eccentricity in the squeeze film at zero shaft speed; the rotor consequently acts as a vertical rotor. The second type is the non-centralised squeeze film which relies on shaft imbalance which, when this force exceeds the gravity load, lifts the main bearing in the squeeze film. Subsequently the dynamic stiffness of the squeeze film results in the lift being maintained. This second form tends to be cheaper and useful when space is restricted. In both types the inner squeeze film surface is prevented from rotating by dogs or a similar device but the inner surface is free to move within the oil film in order to generate the film forces required.

In 1979 Nilolajsen and Holmes (30) experimentally examined a squeeze film sup orting a hydrodynamic bearing and shaft. The separating ring was prevented from rotating and supported by three Belleville washer stacks so that the film could be centralised and the support stiffness could be varied independently to the support damping which was changed by the use of various viscosity oils and squeeze film clearances.

Significant reductions in response amplitudes were obtained by increasing support damping but the critical speed of the system showed little variation with support stiffness in experiments despite theoretical predictions to the contrary. This work shows how attenuation of amplitude of vibration can be achieved by selecting the correct value of support damping as discussed in Gunter's work (13).

The dynamics of a simple single mass rotor rigid shaft with squeeze film supported rolling element bearings was analysed using Ocvirks' solution of Reynolds equation, by Cooper (21). He predicted three orbit solutions for a given unbalance. White (22) experimentally verified the predicted orbits showing the higher eccentricity (whirl) mode and the lower eccentricity (inverted) mode which were stable and the intermediate solution which has always been unstable. The "jump" speeds (speeds at which the bearing tends to jump from the lower eccentricity mode to the upper eccentricity mode) were shown by White to be as predicted by Cooper. However from the correlation of White's experimental and theoretical results it is shown that White's model is only accurate for eccentricity ratio less than 0.4.

SQUEEZE FILM SUPPORTED ROLLING ELEMENT BEARINGS

In the 1970's work was focused on the prevention of large whirl amplitudes exhibited by rotating shafts mounted in rolling element bearings by providing additional support damping in the form of squeeze film surrounding the bearing. The excitation of this system by the rolling elements has been shown by several authors (26, 47) to be negligible.

In 1974 Mohan and Hahn (23) theoretically examined the centrally preloaded squeeze film damper system for rigid rotors to predict the squeeze film forces generated and their effect on system response. Using the short

bearing approximation of Reynolds equation, the film forces and hence the velocity (damping) forces and deflection (stiffness) forces were used to define the linear stiffness and damping coefficients of the film. The model showed the jump from uninverted to inverted mode, predicted by Cooper and experimentally observed by White, and the inverted mode was again reported to give lower vibration amplitudes and transmissibilities. The performance of the squeeze film was shown to be depedant on the absolute values of a particular non-dimensional bearing parameter and on a shaft unbalance parameter. They suggest for low transmissibilities, a low centralising spring rate should be used.

Rabinowitz and Hahn (24, 25, 26) conducted theoretical and experimental invesigations into the stability of a centralised squeeze film mounted flexible rotor with rolling element bearings. They show that the squeeze film pressurization eliminates bi-stable operation, the lower eccentricity mode being adopted, and allows operation through the first pin-pin critical speed which produced violent vibration when the film was underpressurised. Both pressurization (which develops a 2π film) and an increase of the bearing parameter discussed in reference (23) resulted in a more heavily damped system. Fine tuning could be achieved by controlling the lubricant temperature (and hence its viscosity), the control of which was found not to be critical for close to optimum performance. Vibration amplitude at the system critical speed was shown to be attenuated by a factor of up to 25 in some cases. The pressurization levels required for the development of a 2π film were shown to be as had been earlier predicted by Simandiri and Hahn (29).

In 1981 Hahn (27) developed the theoretical model described in reference (23) to allow for pressurization resulting in a full 2π film. The paper provides design data for effective squeeze film performance as vibration isolators. The model suggests that transmissibilities below one can only be achieved in certain cases at the expense of increasing rotor vibrational amplitudes. The bi-stable operation is again prevented by pressurization and with no pressurization there is a critical unbalance above which the squeeze film effectiveness is minimal. In 1976 Gunter, Barrett and Allaire (31) used Reynolds equation to produce film force coefficients for each given bearing position and consequently provide data for optimum damper design in respect of stability and transmissibility. Their model combines the single mass rotor and bearing equations and is integrated forward in time to give system response orbits. Only positive pressures were considered to contribute to film forces. They showed that the 2π film reduces the stiffness of the film to zero but doubles the film damping as compared with that of a cavitated (π) film.

Numerous computer predictions of shaft trajectories were plotted and the model has been applied to several compressors with successful suppression of non-syncronous components of whirl. They conclude that with short rigid rotors, reducing squeeze film clearance will improve stability but for long flexible rotors the same technique can be catastrophic. Increasing the damping and effective shaft stiffness may also improve stability. These conclusions can be related to Gunter's work of 1966 on support stiffness (13) excessive squeeze film stiffness or damping were both shown to reduce damper effectiveness by increasing transmitted forces. Optimum stability and transmissibility values were for eccentricity ratios below 0.4.

Attention was later focused on more realistic rotors by Cookson and Xin Hai Feng (32) who produced an assymetric multi-mass model of a flexible rotor supported by uncentralised and unpressurized squeeze films. This model gave a more realistic representation of a practical rotors system than the usual single mass model. The use of uncentralised squeeze films means that the circular syncronous orbit assumption cannot be used and the linear model restricts its applicabilty to low vibrational amplitudes. The analysis shows that the gravity effects increases the transmissibility of the uncentralised film and that low bearing operating parameter values reduce the effect of foundation stiffness on transmissibility, amplitude ratio and system critical speed. It was shown that the transmitted force could be kept low over the entire speed range, and by varying the support stiffness the systems critical speed could be shifted in the frequency range.

None of the work was experimentally verified and therefore conclusions that have not been previously experimentally confirmed by other authors must be held in question.

Theoretical and experimental work on squeeze films without a centralising spring for both rigid and flexible rotors was published by Cookson and Kossa in 1980 (33). As no film pressurisation was used the π film case applies. The results show as in reference (32) that the gravity parameter should be kept low and that the bearing operating parameter discussed in reference (23) is optimum at 0.1. Instability resulted from lower operating parameter values and the film tended to behave as a rigid support at higher values. The authors state that squeeze films do not reduce vibrational effects by shifting the resonance away from operating speed but that they reduce the amplitude of vibrations by a vibration redistribution process. However this latter conclusion contradicts the theoretical work

presented in reference (32) and later the experimental results shown in reference (47).

From the above studies the ability of the squeeze film to attenuate vibration amplitude and force transmission to supports was shown provided the bearings were not situated at rotor free-free nodes and that the film stiffness was not such as to promote instability. The majority of theoretical models showed good agreement with experimental results in the lowe eccentricity rnage but tended to give poor prediction accuracy at high (E>0.4) eccentricity ratios. Work by Bannister (15) in 1972 on the non-linear characteristics of hydrodynamic film forces, showed the excessive inaccuracies of linear models for E>0.4. By developing the second differential of the fluid film forces with respect to deflection, (ignored by linear models) he produced 28 coefficients to model the film and with experimental support, showed that for amplitudes above 15% of radial clearance greatly improved predictions result.

Despite the accepted predominance of non-linear oil film forces at high eccentricity ratios, the simplification of the analysis with the consequent reduction in computer time required still leads many investigators to adopt linear models. If the orbit amplitudes are kept low these models are sufficiently accurate to justify the simplification.

HYDROSTATIC BEARINGS

The static properties of hydrostatic bearings are well documented by authors Raimond and Boyd (36) and with additional design data by O'Donoghue and Rowe (35,46). These cover investigations into various bearing geometries with effects on static loads bearing stiffnes, oil flow rate etc, usually the restrictors used are of the capillary or orifice design.

In 1976 Singh and Singh (37) developed a method of stiffness optimisation of a hydrostatic bearing by the use of spring loaded piston type variable restrictor. This was similar in operating principle to the spring loaded diagphram, where the recess back pressure forces the piston back resulting in a reduced resistance to flow and an increase in flow rate and fluid pressure. Unfortunately Singh gave no static or dynamic performance data for his design but did provide a useful demonstration of how restrictor design can take into account bearing parameters to give the required stiffness.

The papers dealing with the dynamic characteristics of hydrostatic bearings are scarce presumably due to the, to date, satisfactory dynamic performance of hydrostatic bearings based on static considerations. However significant contributions have been made by the following authors.

Licht and Cooley (38) theoreticlaly studied the dynamic performance of a hydrostatic bearing and concluded that, for incompressible flow, large squeeze numbers (the ratio of volume flow rate due to vibration to static volume flow rate) gives high damping and low squeeze numbers gives low damping. Unfortunately the effects of restrictor type or vibrational frequency are not considered.

Ingert (39) in 1973 developed a bearing characteristic equation and used its roots to define the limits of stability. He concludes that for realistic pad geometries instability is unlikely and that high oil viscosities, high bearing area and and low bearing clearance, and for compressible flow, higher pocket pressure results in reduced vibrational amplitude. Each of these factors leads to higher bearing stiffness.

It was shown by Newton and Howarth in 1974 (4) that the tilt and dynamic action of a hydrostatic bearing would change the static operating variables and conclude that static stiffness and damping are highest for no tilt. Unfortunately the model does not consider dynamic stiffness and damping and ignores the effect of inertia by the use of steady velocities. Cooms and Dowson in 1965 (47) published work showing that for static considerations inertia effects were insignificant except at very low supply pressures. The importance of this work is that it provided qualitive data to support the assumptions made in most investigations of the static properties of hydrostatic bearings, that the inertia effects can be neglected. However, the effect of inertia on the dynamic properties of a hydrostatic bearing with a relatively large restrictor fluid volume can still be considered significant (43).

Brown in 1961 (41) analyses the dynamic behaviour of hydrostatic bearings by assuming compressible laminar flow and neglecting fluid inertia. He represents the bearing system by a series damper and spring to represent the dynamic stiffness and damping with a parallel spring to represent the static stiffness, hence assuming that they are independent. He suggests that the dynamic properties were largely dependent upon the fluid under the sills, i.e. the fluid in the pocket and between the pocket and restrictor could be neglected when deriving compressibility and that the damping was independent of inlet restrictor. These assumptions, although shown to be invalid (48) gave good results up to 20 Hz but large errors above this frequency. He concludes that dynamic stiffness is always higher than static stiffness and that damping is always high.

HYDROSTATIC SQUEEZE FILM

Due to the need for stability of hydrostatic journal bearings of higher opearating speed, Choy and Halloran in 1982 (42) conducted a theoretical and experimental investigation into the damping and stabilising effects of a hydrostatic journal bearing. The linear damping forces are calculated using Reynolds equation and the equilibrium positon found by iteration of pocket pressure by equating the inlet and leakage flow of each pocket. The film showed very little change in stiffness or damping with supply pressure increase but both the stiffness and damping of the film changed significantly with eccentricity variation; the maximum values occuring higher eccentricity levels for larger film clearances but the peak values reducing in amplitude as clearance increased. The damping is shown to be due to the fact that the oil must pass over the sill and is not dependent on the amount of flow to the recess.

Koshal and Rowe in 1981 (44,45) experimentally and theoretically investigated the hybrid operation of hydrostatic bearings and confirmed the benefits of high load capacity at zero speed and high stiffness for low eccentricity operation, attributable to hydrostatic action and high stiffness and load capacity at high eccentricity, attributable to hydrodynamic action. The experimental runs shown a possibility (at high power rates, frictional power/pumping power, i.e. E>0.75) of negative bearing stiffness (oil film collapse). They suggest that this is due to high film thickness gradients at high eccentricity ratios with the associated circumferential flow and severe circumferential pressure gradients with the resultant reduced effective high pressure area. The increase in slot pressure as eccentricity tends to one cannot compensate for the loss in effective high pressure area and the bearing collapses.

As the phenomena was never experienced during their experiments, it is hard to know whether the explanation is correct or indeed whether the phenomena exists.

VARIABLE IMPEDANCE BEARINGS

The dynamic behaviour of a shaft bearing system, as with all mechanical systems, is determined by the mechanical impedances or more accurately the interaction of the mechanical impedances of the system. These impedances can be represented in complex form with the intertias and stiffnesses represented by opposing imaginary components and the damping represented by the real component. The impedances are a function of frequency and at certain frequencies the imaginary components cancel out and system instability results.

The above suggests that the dynamic behaviour of a rotating shaft system can be controlled by the control of the system impedances. Due to the limitations put on turbine designers, the mass, internal damping and stiffness of the components are generally fixed for a particular system, the support structure being the only component available for modification of impedance values.

Goodwin in 1981 (43) experimentally and theoretically investigated a hydrodynamic journal bearing surrounded by an outer hydrostatic bearing which was pinned to prevent rotation. The hydrodynamic film was modelled using eight linear film coefficients and the effect of variation of the bearing parameters on the system performance was investigated. The theory and experimental results show the ability to independently control the hydrostatic static stiffness by altering the setting of a flow restrictor in the hydrostatic film supply line, whilst the X and Y axis dynamic

stiffness and damping can be controlled by varying the flow resistance of a restrictor which linked the hydrostatic bearing pockets to an accumulator. Substantial variations of the overall system stiffness and damping were recorded due to the introduction of hydrostatic film. These were measured at various excitation frequencies. Of particular reference was the shaft of system critical speed by 350 rpm and the reduction of vibrational amplitude by 80 per cent predicted for a large machine when operated with hydrostatic supports.

Burrows, Sahinkaya and Turkay (49) in 1984 carried out theoretical and experimental investigations into the effect of optimising the damping for a given support stiffness and shaft speed. The damping was introduced by a squeeze film surrounding a rolling element bearing and controlled by applying a step change in the oil supply pressure. As shown by (42) no damping changes occur with supply pressure increase unless the pressure increase is supressing the onset of cavitation in the squeeze film. Presumably this control of cavitation is the mechanism being utilised in this investigation. They state in conclusion that for the conditions of their experiments the adaptive dampers had little effect on amplitude compared with passive dampers when the squeeze films were situated at the shaft ends and that the transmitted force increased.

They do however display the beneficial effect of Belleville washers in changing the support stiffness and being able to 'tune' the system support damping for a given support stiffness which together modify the critical speed and response amplitude.

In 1979 Sandler (48) investigated the automatic control of the dynamic response of a rotating shaft system. The experiments were based on the variation of the shaft stiffness by automatically controlling the bearing position on the shaft. Two bearing positions were possible each with an associated resonance peak for each critical speed. The result of changing the bearing position during speed increase or decrease was that the system adopted the response of the associated bearing position i.e. the lower of the two dynamic responses. Despite the obvious limitations of changing shaft stiffness on a practical system, the work does show how automatic control of system impedances, in this case shaft stiffness, can be utilised to modify the dynamic behaviour of a system.

The increase in power output of turbine systems over the past thirty years has been accompanied by higher running speeds and rotor length to diameter ratios, with the result that systems often operate above several of their critical speeds. Because of the difficulty in accurately predicting support dynamic impedances and rotor dynamic behaviour, rotor system designers sometimes find predictions of rotor system critical frequencies sufficiently inaccurate as to result in the operating speed coinciding with or close to the system critical speed. This has resulted in high vibrational amplitudes and transmitted forces at the normal operating speed.

The ability to control the support impedance of the shaft system after the system has been installed will result in a control of system critical frequencies with the consequent ability to shift them away from operating speeds and the resultant reduction in response amplitudes. For system speed run up or for a variable speed system the ability to control the support impedance (particularly stiffness) over a range of values depending on shaft speed, makes available the possibility of a speed controlled

automatic system which will produce low response amplitudes at any running speed.

The possibility of variable impedance support control system failure must also be considered during the development of the system.

STATIC CHARACTERISTICS

SINGLE HYDROSTATIC PAD AND FULL HYDROSTATIC RING

STATIC CHARACTERISTICS OF A SINGLE CURVED HYDROSTATIC PAD

INTRODUCTION

The proposed solution to the requirement for a variable impedance support for turbine systems is based on a modified hydrostatic bearing system. The main bearing, required to accomadate shaft rotation, will be a rolling element type, with the outer surface being prevented from rotating and acting as one surface of the hydrostatic pads. The bearing outer surface will however be free to move within the hydrostatic support clearance. As a result of this relative motion, together with a continuous oil supply to the clearance, the land areas of the hydrostatic support will act a partial squeeze film, sometimes used in its full form to introduce system damping. The hydrostatic effect will supply the static stiffness to oppose the gravity load of the shaft system and centralise the shaft, this is normally supplied by a form of mechanical centralising spring with the full squeeze film systems. As a result of this inherent centralising stiffness from the hydrostatic effect, the design will possess an oil filled recess. The presence of this recess results in reduced land area and hence reduced sqeeze film forces but also introduces the possibility of varying the dynamic stiffness of the support independantly to the static stiffness.

The dynamic characteristics of a hydrostatic system are dominated by the effect of flow variations imposed by the dynamic load. When the load approaches the hydrostatic pad, the recess pressure will increase and the supply flow reduce and in certain cases reverse. The flow variations can be controlled by incorporating an additional exit for the recess oil. An accumulator can therefore be introduced into the system, the effect of which can be varied by varying the value of the assosiated accumulator restrictor. With this modification, as the main bearing outer shell approaches the hydrostatic recess, the recess oil will be partially compressed and partially accelerated to flow either over the lands, down

the supply restrictor or down the accumulator restrictor. It will therefore be possible to control this oil flow and consequently modify the supports displacement opposing forces (stiffness) due to recess fluid compressibility and the velocity opposing forces (damping) thus meeting the requirement for a variable impedance support to modify the rotor systems dynamic response.

SINGLE PAD ANALYSIS

The analysis that follows is in reference to a single curved hydrostatic pad. The responses given are those due to vertical deflection of the supported load with velocity and acceleration both zero. The dimensions and other pad values are similar to those of practical use for a 1/20 model of a typical 750 kN rotor system supported with one pad at each end, the theory is however applicable to any pad dimensions with any exceptions commented on in the text.

To allow simplification of the analysis, the leakage pressure of the pad, i.e. the fluid pressure at the land outer perimeter, is assumed zero (relative to atmospheric pressure). When the multi pad system is analysed this enables inter pad flow and other pad interactions to be neglected. The practical realisation of no inter pad flow is achieved by drainage channels running along the axial length of the hydrostatic ring between each of the pads.

The system can be considered as one fixed and one variable resistance representing the input restrictor and the parallel plate land resistance respectively. A static supply pressure causes a flow through tl . pad the magnitude of which is controlled by the two resistance values. The pressure drop across the input restrictor is directly proportional to the volumetric flow rate and thus determines the pocket pressure which opposes

the load. The parallel plate resistance is proportional to the inverse of the land clearance cubed which in series with the restrictor resistance determines the flow rate. Consequently by conservation of flow the pocket pressure can be determined for a given pad deflection.

The electrical analogy of the system is a supply voltage, a fixed resistance, a load voltage and a series resistance α 1/ charge^3, the current representing the volume flow rate. Using this it can be seen Fig. 1 that the resistances are acting as dampers in the system i.e. generating a back pressure proportional to the flow rate. Despite this damping system the overall effect is to change the pocket pressure which will be 'seen' as a stiffness by the applied load. This apparent 90^o phase shift of the fluid flow damping to generate a pad stiffness effect (Fig.1) is due to the steady fluid flow required to support the load. Thus this fluid flow damping can be substituted by a spring in the mechanical analogy whose stiffness is constant (linear model) or $f(\epsilon)$ non-linear model. It should be noted that the fluid compressibility, system resistance and fluid inertia do not undergo similar 90^o phase shifts for the dynamic model and remain stiffness, damping and inertia respectively.

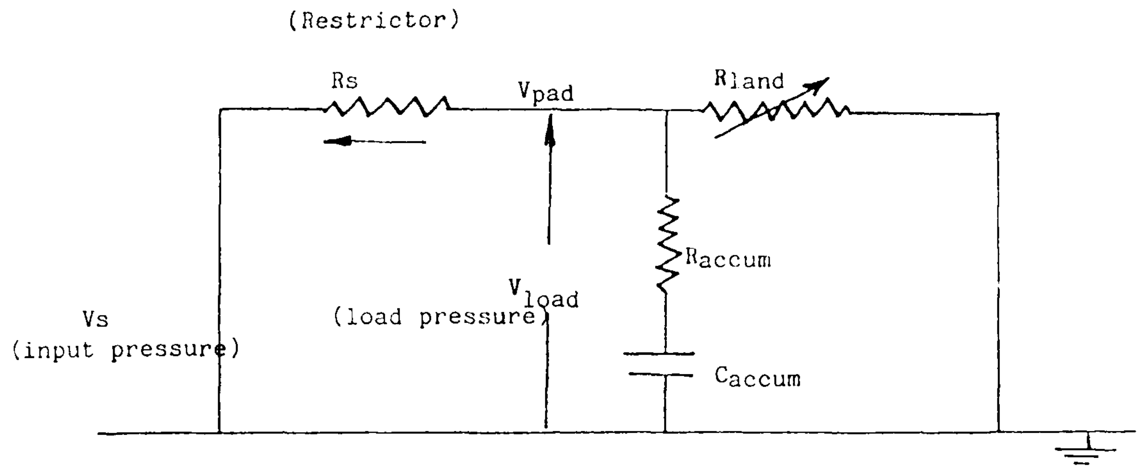
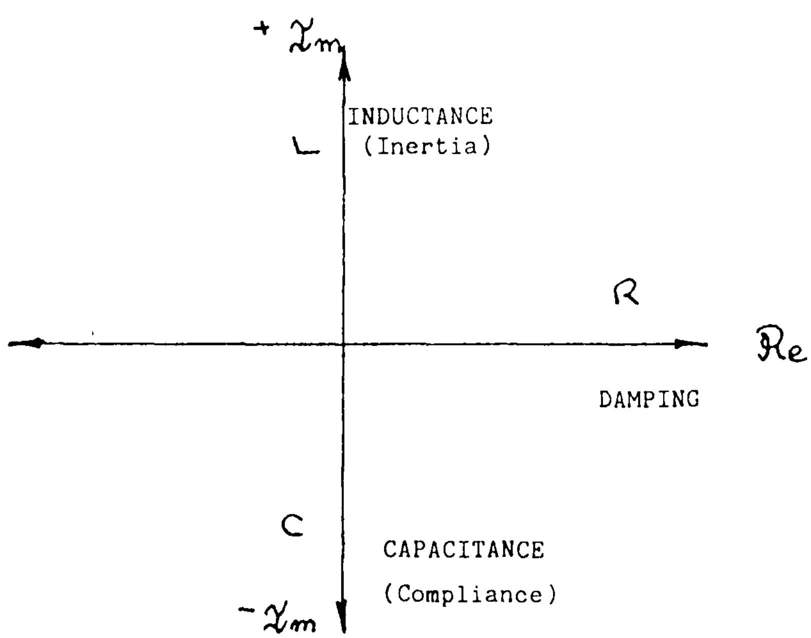


Fig. 1

ELECTRICAL ANALOGY
FOR STATIC LOAD



COMPLEX PLANE

MIDLAND APPROXIMATION

The land parallel plate resistance causes a linear pressure drop with distance across the land. as the pressure drop is linear then this pressure gradient can be substituted for a step drop in pressure at the end of a land of half the original width. Thus this midland approximation can be used to simplify the complex pressure profile of the hydrostatic pad. Goodwin (42) shows that for a pad recess pressures of 10 Pa and a consequent pad load of 17.7×10^{-3} N an error of 1.7% above true load is calculated using this midland approximation. Due to the more rapid pressure drop with distance at the land corners the midland accuracy will be dependent on the percentage of land corner area to straight land area, the example given by Goodwin being for a percentage corner to total land are of 22%. Any significant increase above this value may require a correction factor for the final model.

DISCUSSION

The graph of load v eccentricity (appendix 1) shows the trend for pad load to tend to a maximum value of (Ps xAe) as eccentricity reduces to $^-1.0$ for all B values. This trend can be explained by referring to the mechanics of the pad system.

As eccentricity approaches $^-1.0$ the land clearance tends to zero and the associated resistance tends to infinity. The resultant zero flow rate causes the pressure drop across the inflow restrictor to become zero and the pad pressure tends to the supply pressure independent of inflow resistance and therefore β value.

As the eccentricity increases above $\epsilon = {}^{-}1.0$ the land resistance assumes a finite value and flow recommences. The rate at which the pressure drop across the restrictor increases with flow rate increase is by definition the restrictor resistance value, thus the higher the restrictor value the faster the pressure drop increases with flow. This is shown by graph 1 with the lower β values displaying higher load eccentricity gradients.

The stiffness of the pad is defined by the rate of change of load with deflection of the shaft which is given by the gradient of the load eccentricity graphs. As shown by graph 2, the magnitude of maximum stiffness and the eccentricity at which it occurs increase with β ratio reduction, these maximum stiffness values represent the points of inflection of the load curves. The lowest stiffness values occur at an eccentricity value of $\epsilon = {}^{-}1.0$ due to the asymptotic approach of the load curves to the limiting value. Work by Burrows, Sahinkaya and Turkay (49) and Goodwin (42) show that the first and second critical speeds of a typical system are most sensitive to support stiffness changes when the support stiffness is of the order of 5×10^4 NM^{-1} and that the third and fourth critical speeds are most sensitive to support stiffness changes when the support stiffness is of the order of 1×10^6 NM^{-1}. Thus a range from 5×10^4 to 1×10^6 NM^{-1} would be desirable in shifting the critical speeds of the system away from running speed.

To achieve this low value, the stiffness graph suggests operating at high β values (approaching 1.0) and eccentricities below $\epsilon = {}^{-}0.9$ or lower β values (below 0.1) and eccentricities above $\epsilon = 0.9$.

Adopting the high β solution would require high static deflections which may impose unacceptably high stress levels on the rotor system due to shaft misalignment. Uncentralised squeeze films do however successfully operate at similarly high eccentricity ratios particularly at the low speed range which suggests that the system could accommodate such high eccentricity values. However, referring to graph 1, this shows that this would require a reduction in supply pressure to give the required working load which would result in no overload capability. Using $\beta = 0.1$ necessitiates eccentricity values of $\epsilon = >+0.9$. Referring to graph 1 the supply pressure increase required would result in unacceptably high flow rates. The pad stiffness is also proportional to supply pressure and consequently an increase in supply pressure would also result in a corresponding stiffness increase thus the low β solution is also unacceptable.

Referring to equation (7) the final possibilities is an increase of the land clearance at $\epsilon = 0$. This however both increases the flow rate (graph 4) and the deflection required before load increases are opposed by the pad, consequently shaft stresses would increase. Increase of the clearance does however reduce the pad stiffness by reducing the rate of flow change w.r.t. deflection. Unfortunately (see graph 4) the associated flow increase is severe enough to make clearance increase unsuitable as a means of reducing pad stiffness.

The above analysis shows that modification of standard hydrostatic pad parameters is not a suitable method of reducing support stiffness to the value required and that the method of accumulator installation must be investigated in order to change the dynamic stiffness independently of the static stiffness.

The full hydrostatic support consists of four pads at 90 degree intervals around the shaft axis and in the same plane as the main shaft bearings. Drainage channels at atmospheric pressure run axially between each pad and prevent pad flow interaction. The characteristics of each pad can therefore be calculated individually, with consideration given to the different relative shaft position, and the results directly added together.

As the pads are identical and the pad system symetrical about both the X and Y axis, the graphs for the four pad system show the same symetry (Graphs 6-11). The pad reaction force decreases with shaft distance from the pad, however the opposing pad force will increase until at the shaft central position, the two reactions will cancel giving a zero resultant force. As a result of this, despite the variations in pad normal reaction force as the shaft moves across the pad face, the resultant force remains zero (Graph 10). This is not the case for the pad system stiffness profile (Graph 9). The static stiffnesses of the pads are always positive and the summation of such stiffnesses always gives a positive resultant. For $\beta \geqslant 0.5$ the maximum stiffness accurs at the central shaft position (Graph 7) due to the minimal peak and gradual stiffness changes with normal shaft deflections (Graph 2). For lower β values, the maximum stiffness peak for the single pad (Graph 2) is severe enough to remain when the opposing pad effects are added.

Despite the lack of reaction force from the land surfaces, no accute reduction in reaction force accurs when the shaft is loaded in a direction along the shaft centre to drainage channel centre line with the pad dimensions used (Graph 13). This suggests that no extreme radial static stiffness variations will occur as the shaft travels around the hydrostatic clearance circle due to shaft unbalance, even if eccentricities are close to unity. This situation must be reassessed for the final pad dimensions to

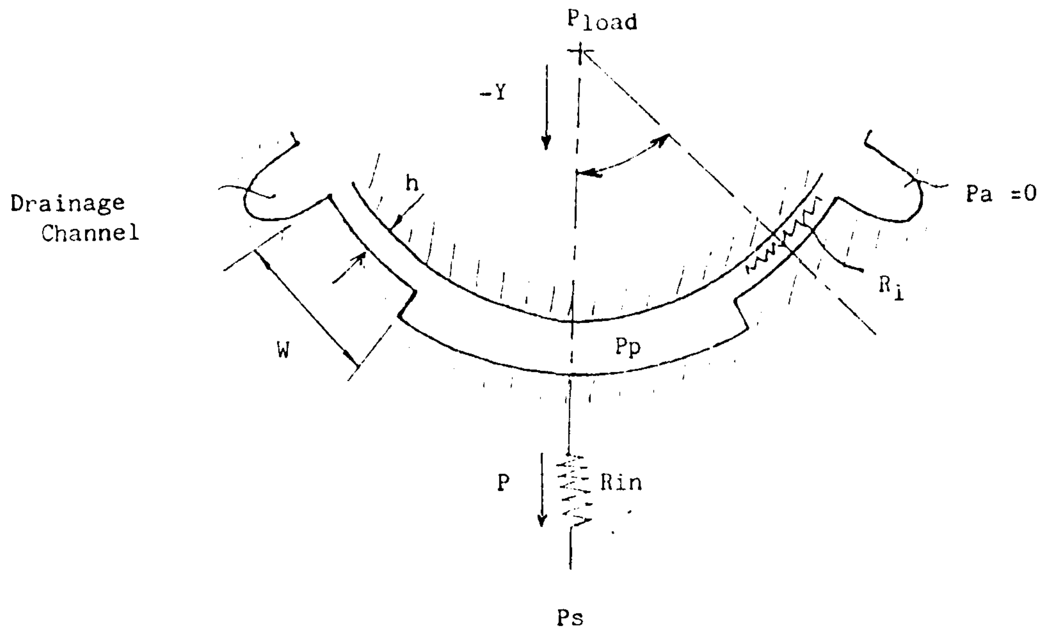
ensure that minimal angular variations still exist.

Graph 12 shows the cross coupled stiffnesses for the four pad system ie. the effect on stiffness characteristics in one direction due to shaft motion in the normal direction in the bearing plane. As shown, the cross coupled effects remain zero when the shaft is on axis X or Y. This simplifies the situation of a static load acting through the lower pad upper pad centre line. However, the off axis cross couplings are non zero and must therefore be taken into account if either the static loading is not vertical or if the position of the shaft changes in the bearing plane due to unbalance forces lifting and orbiting the main bearings.

The flow rates of the total support are simply the sum of the individual pads calculated with the shaft in its correct poition relative to the pad. The shaft centred flow rate is therefore four times the single pad ϵ=0 flow rate (Graph 3 & 11). For β=0.9 as the shaft shaft moves from ϵ=0 away from the pad, the flow rate increases far more sharply than the flow from the oppisit pad decreases, consequently the flow rate increases with eccentricity ratio for high β values. For β=0.1 the flow variation with eccentricity is very small and consequently the sum of the opposing pad flows remains almost constant. Obviously the magnetude of the high β value flows are higher due to the higher pocket pressure for a given eccentricity. Due to the predictability of the flow patterns no problems should occur as a result of excessive support flow rates.

The above analysis is not altered by the additon of an accumulator and the assosiated resistance Raccum (Fig.1) regardless of the accumulator operating parameter B= ΔP/ ΔVol. The flow into the accumulator will stop when the accumulator pressure equals the pocket pressure. Thus the static stiffness, load and flow remain unaltered. The B value can be assumed zero

for small amplitude dynamic variations but obviously has a finite . It can therefore be represented by a capacitor in the electrical network Fig 1. With a change in load voltage this capacitor charges but the network is subsequently unaffected.



SINGLE CURVED HYDROSTATIC PAD

NOTATION

α =	Midland to midpad angle	Degrees
h =	Land clearance	M
c =	Design clearance @ y=0	M
y =	Shaft surface deflection	M
ε =	Shaft eccentricity ratio y/c	
W =	Land width	M
Ae =	Projected effective area of pad	M^2
L =	Midland perimeter	M
K =	Inverse capillary resistance 1/Rin	$M^5 N^{-1} S^{-1}$
Ps =	Supply pressure	NM^{-1} (Pa)
Pp =	Pad recess pressure	NM^{-1} (Pa)
β =	Pp/Ps	
μ =	Oil dynamic viscosity	Pa.S
Kyy =	Bearing stiffness (vertical)..	NM^{-1}
Kxx =	Bearing stiffness (horizontal)..	NM^{-1}
Q =	Oil flow rate	$M^3 S^{-1}$
P =	Pumping power	W
F =	Bearing load	N

ΔP = Pressure drop across Rin = Rin . Qin

$$\rightarrow Q_{in} = \frac{\Delta P}{R_{in}}$$

$$Q_{in} = K\,(P_s - P_p) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

Land resistance $R_l = \frac{12\mu W}{h^3\ L}$

ΔP_l = Pressure drop across land = $R_l . Q_{out}$

$$\rightarrow Q_{out} = \frac{\Delta P_l}{R_l} = \frac{h^3 L}{12\mu W}(P_p - P_a)$$

$$Q_{out} = \frac{h^3 L}{12\mu^W} P_p \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2)$$

Flow conserved

Therefore $K\,(P_s - P_p) = \frac{h^3 L}{12\mu W} P_p \quad \ldots\ldots\ldots\ldots\ldots\ldots (3)$

$\beta = P_p/P_s$

$$\rightarrow K = P_s(1 - \beta) = \frac{h^3 L}{12\mu W}\beta Ps$$

$$K = \frac{\beta}{1-\beta}\ \frac{h^3 L}{12\mu W}$$

$$K = \frac{\beta}{1-\beta}\ \frac{c^3 L}{12\mu W} \qquad @\ y = 0 \quad \ldots\ldots\ldots\ldots\ldots\ldots (4)$$

Input restrictor value can be calculated for required pressure ratio @ y = 0

From (3) $K(P_s - P_p) = \frac{h^3 L}{12\mu W} P_p$

$$= [(1-\epsilon)c]^3\ \frac{L}{12\mu W}\ P_p$$

Substitute (4) $\frac{\beta}{1-\beta}\ \frac{c^3 L}{12\mu W}\ (P_s - P_p) = \frac{(1-\epsilon)^3\ c^3\ L\ P_p}{12\mu W}$

$$\frac{\beta}{1-\beta}(P_s - P_p) = (1 - \epsilon)^3\ P_p$$

$$P_s = P_p\ (1-\epsilon)^3\ \frac{(1 - \beta)}{\beta} + P_p$$

$$P_p = \frac{P_s}{1 + (1-\epsilon)^3 \frac{(1-\beta)}{\beta}} \quad \ldots\ldots(5)$$

$$A_e = 2(L - W).R.\sin\alpha$$

$A_e =$

$$F_y = \frac{Ps.\ Ae}{1 + (1-\epsilon)^3 \frac{(1-\beta)}{\beta}} \quad \ldots\ldots(6)$$

$$\epsilon = y/c$$

Therefore $F_y = \dfrac{Ps.\ Ae}{1 + (1 - y/c)^3 \frac{(1-\beta)}{\beta}}$

Sideways Pad Force Assumed Zero Therefore $F_{xx} = 0$

$K = \dfrac{dF}{dy}$ Therefore $K_{xx} = 0$

$$K_{yy} = \frac{-Ps.Ae \frac{(1-\beta)}{\beta} .3.\ (1 - y/c)^2\ (-1/c)}{[(1-y/c)^3.\ \frac{(1-\beta)}{\beta} + 1]^2}$$

$$K_{yy} = \frac{-\ Ps.\ Ae.\ \frac{(1-\beta)}{\beta}\ .\ 3.\ (1-\epsilon)^2\ (-1/c)}{[(1-\epsilon)^3\ \frac{(1-\beta)}{\beta} + 1]^2} \quad \ldots\ldots(7)$$

Volumetric Flow Rate Q

From (2)

$$Q = \frac{h^3 L}{12\mu W} .P_p$$

$$Q = \frac{[1-\epsilon)^3\ c^3 L]}{[\ 12\mu W\ ]} \left[\frac{Ps}{(1-\epsilon)^3 \frac{(1-\beta)}{\beta} + 1} \right] \quad \ldots\ldots(8)$$

Pumping Power P

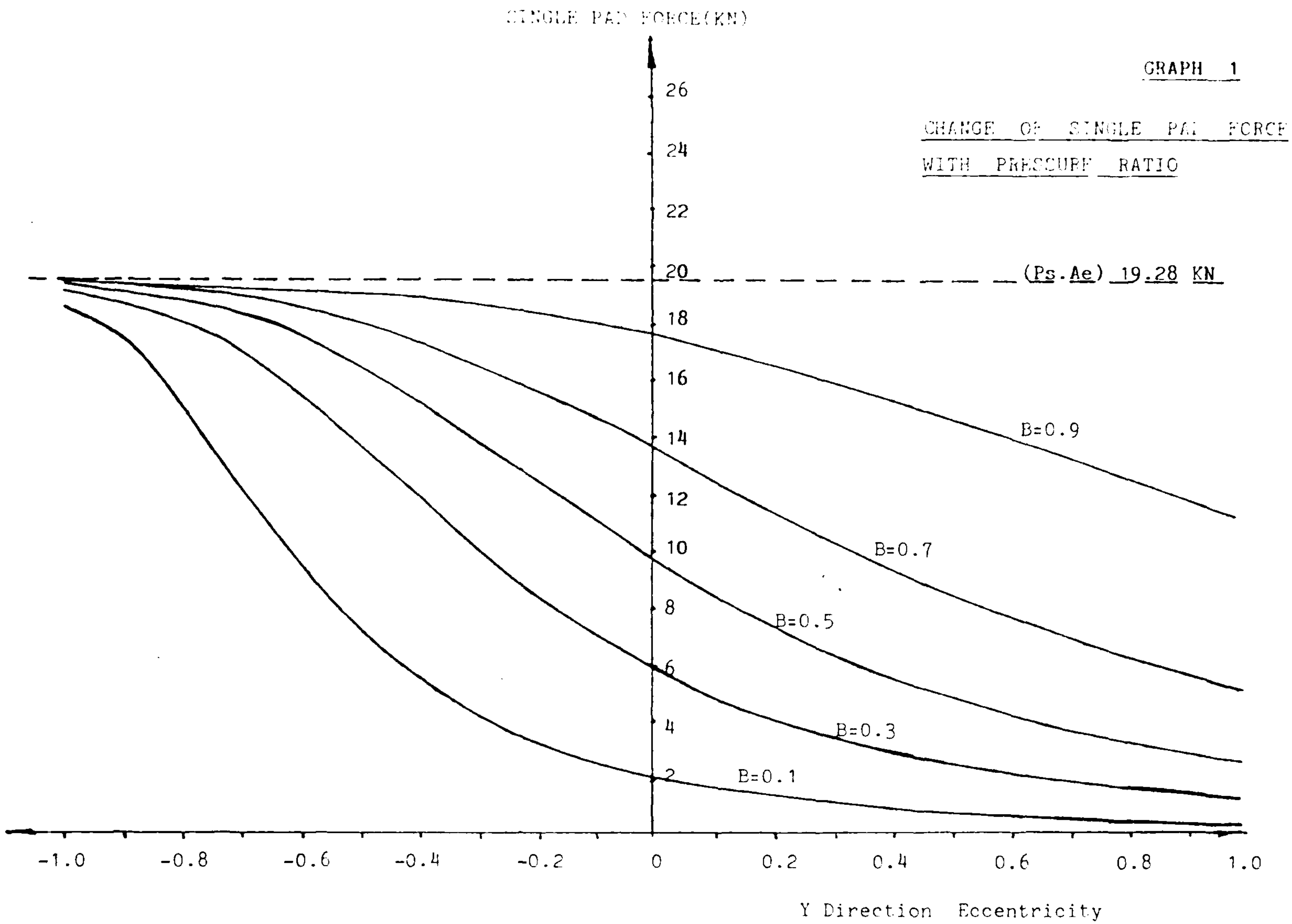
$$P = Q.\ Ps$$

$$P = \left[\frac{(1-\epsilon)^3\ c^3 L}{12\ \mu W}\right]\left[\frac{Ps}{(1-\epsilon)^3\ \frac{(1-\beta)}{\beta} + 1}\right] \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ (9)$$

The above was converted into computer program and the results plotted. See Appendix.

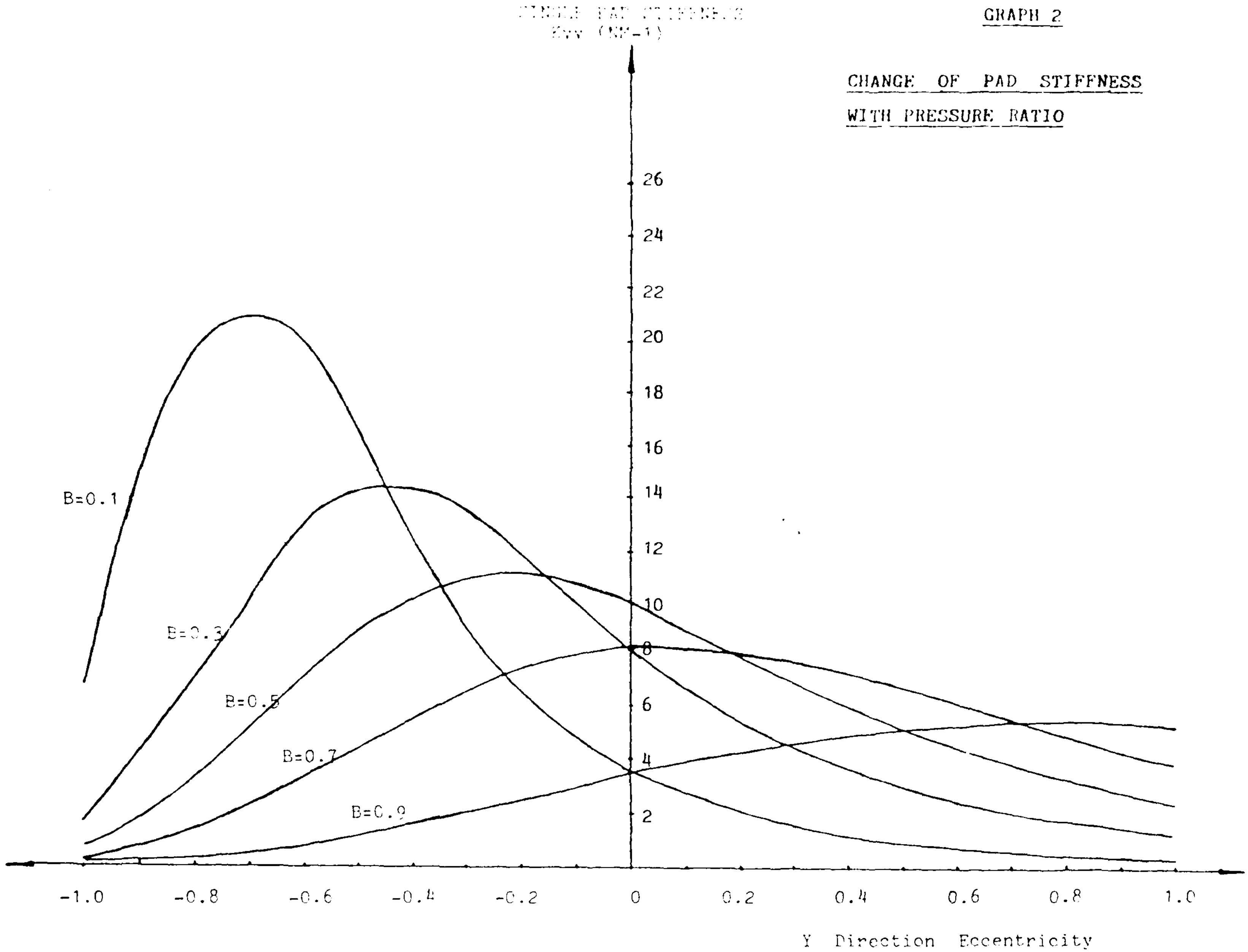
GRAPH 1

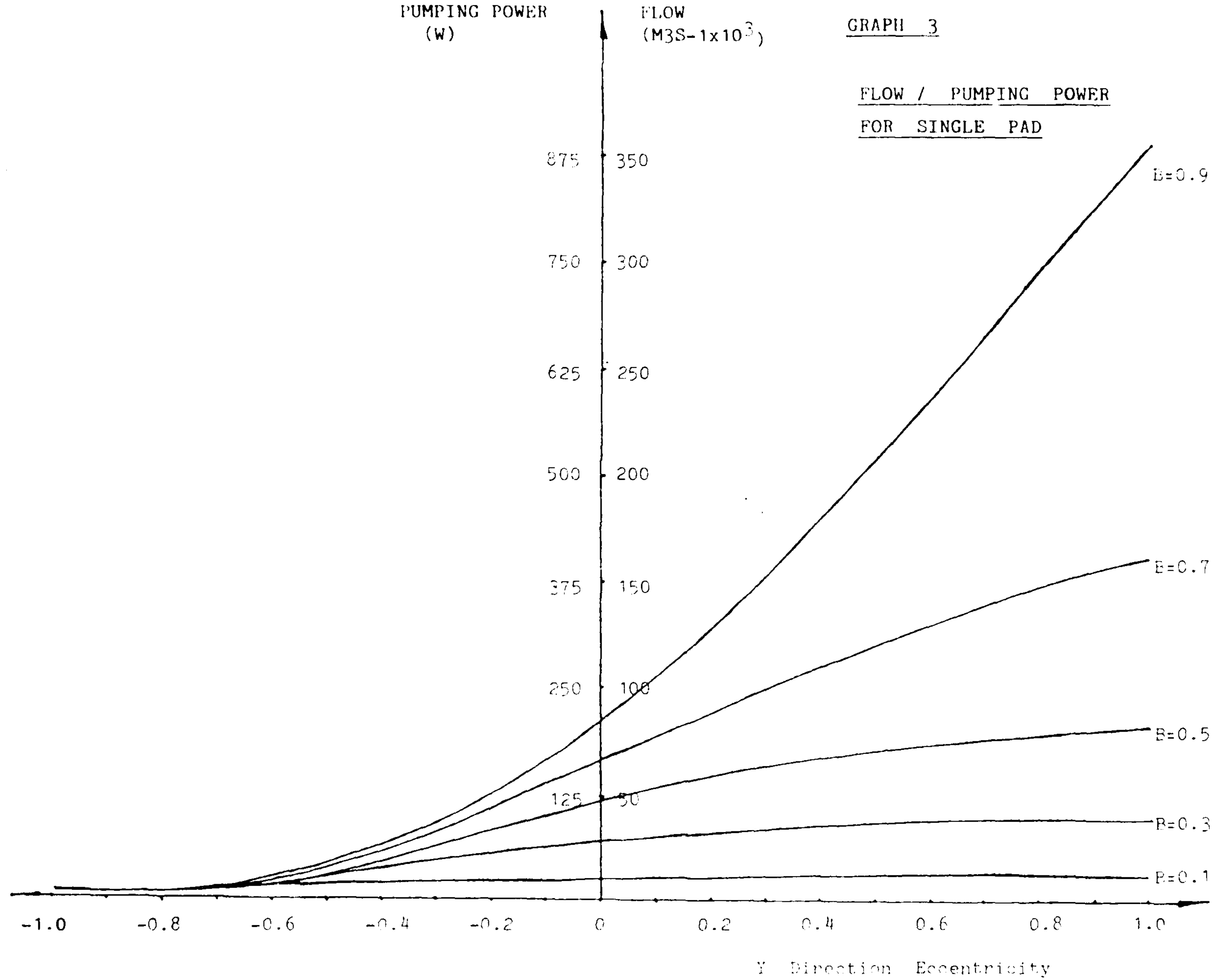
CHANGE OF SINGLE PAD FORCE WITH PRESSURE RATIO



GRAPH 2

CHANGE OF PAD STIFFNESS WITH PRESSURE RATIO



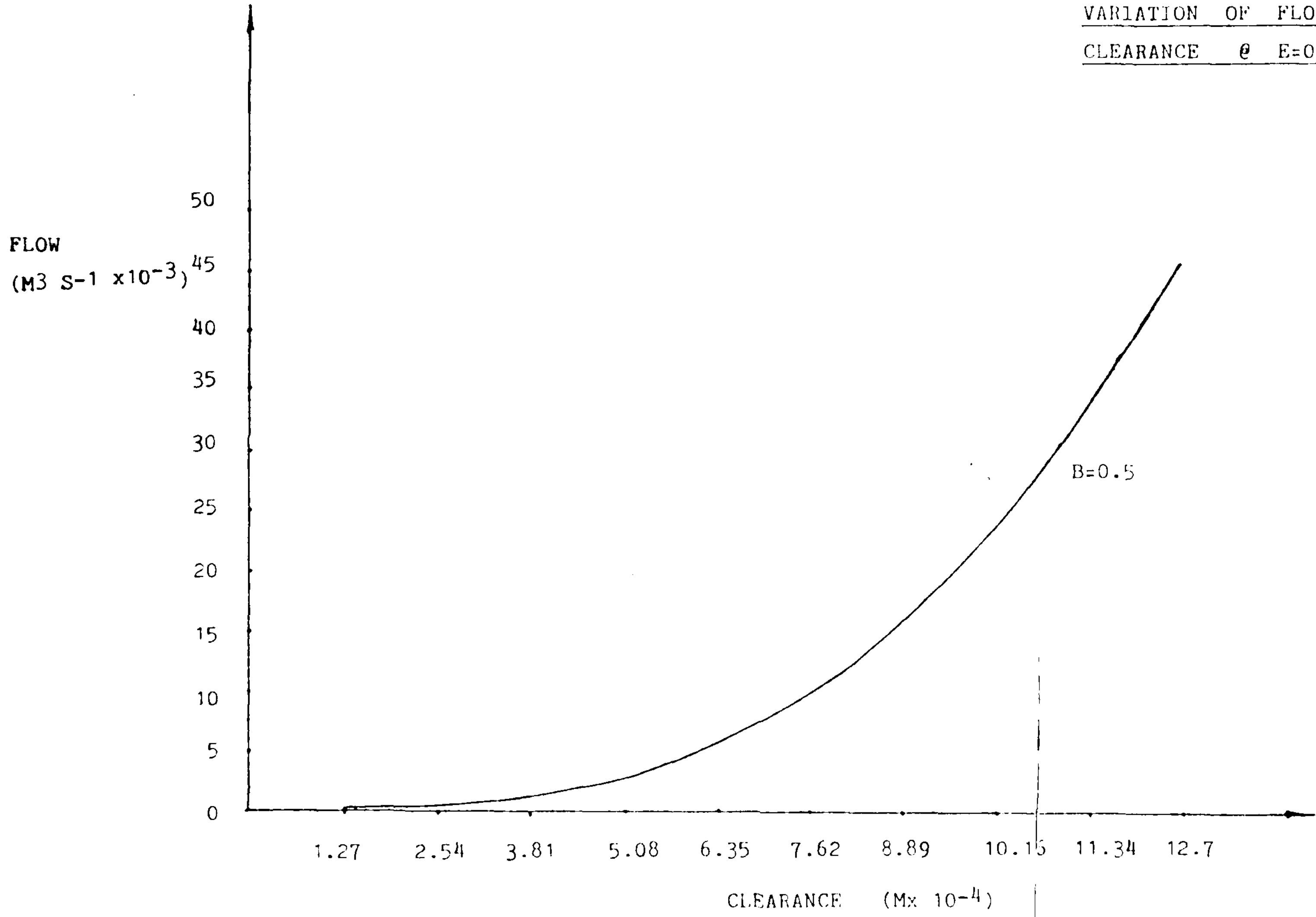


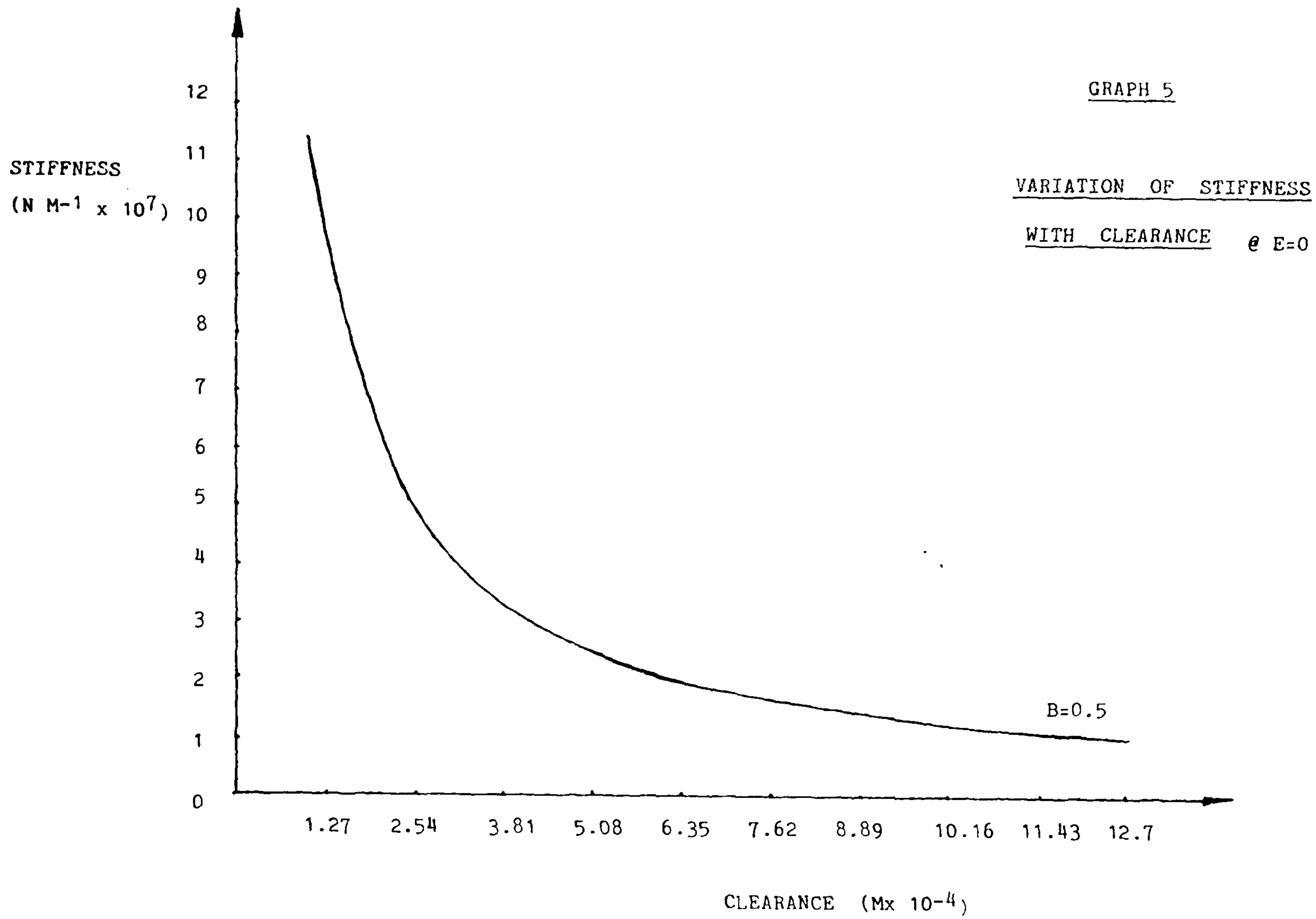
GRAPH 3

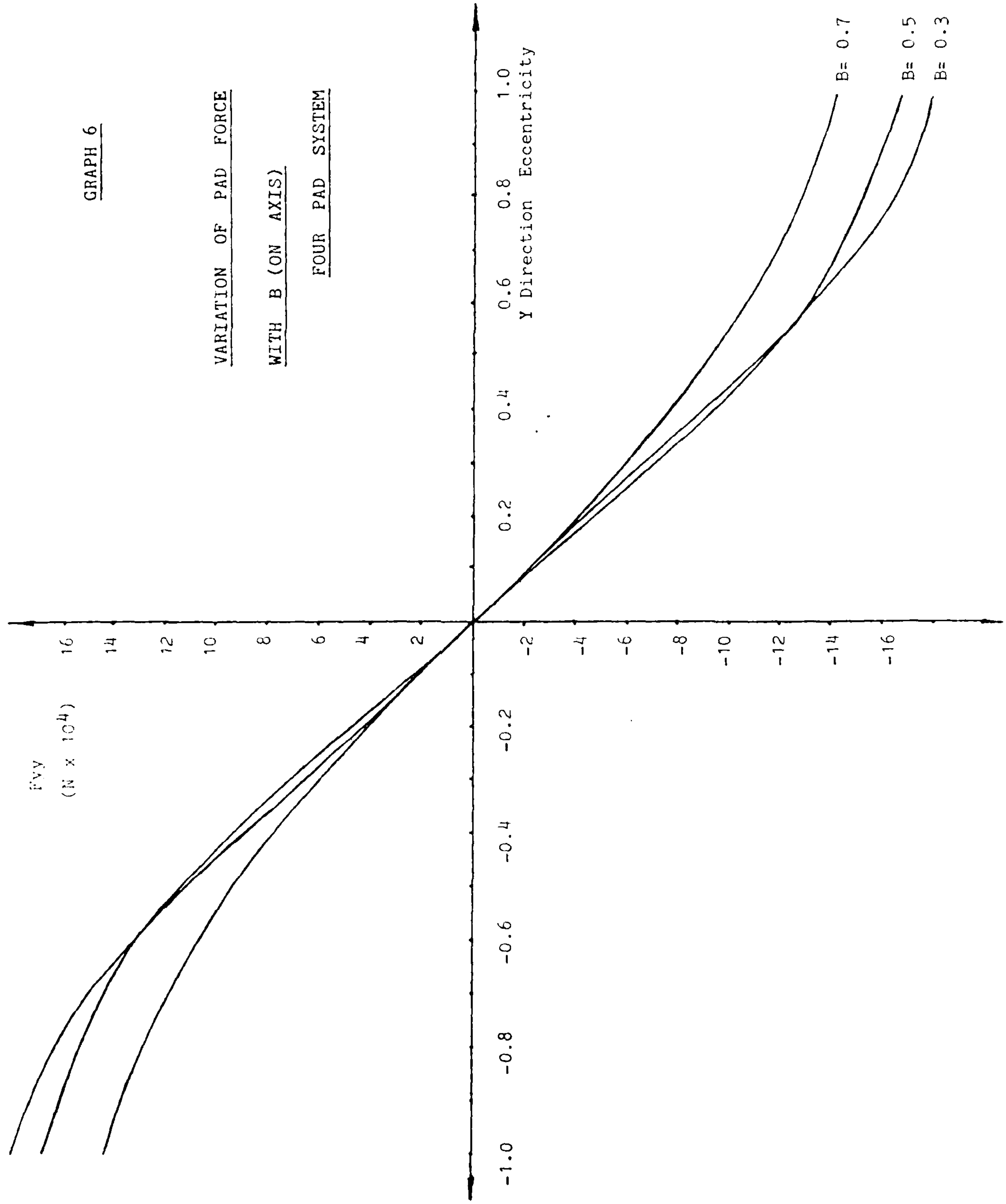
FLOW / PUMPING POWER FOR SINGLE PAD

GRAPH 4

VARIATION OF FLOW WITH CLEARANCE @ E=0




GRAPH 5
VARIATION OF STIFFNESS
WITH CLEARANCE @ E=0
STIFFNESS
(N M-1 x 10^7)
12
11
10
9
8
7
6
5
4
3
2
1
0
B=0.5
1.27
2.54
3.81
5.08
6.35
7.62
8.89
10.16
11.43
12.7
CLEARANCE (Mx 10-4)



GRAPH 6

VARIATION OF PAD FORCE WITH B (ON AXIS)

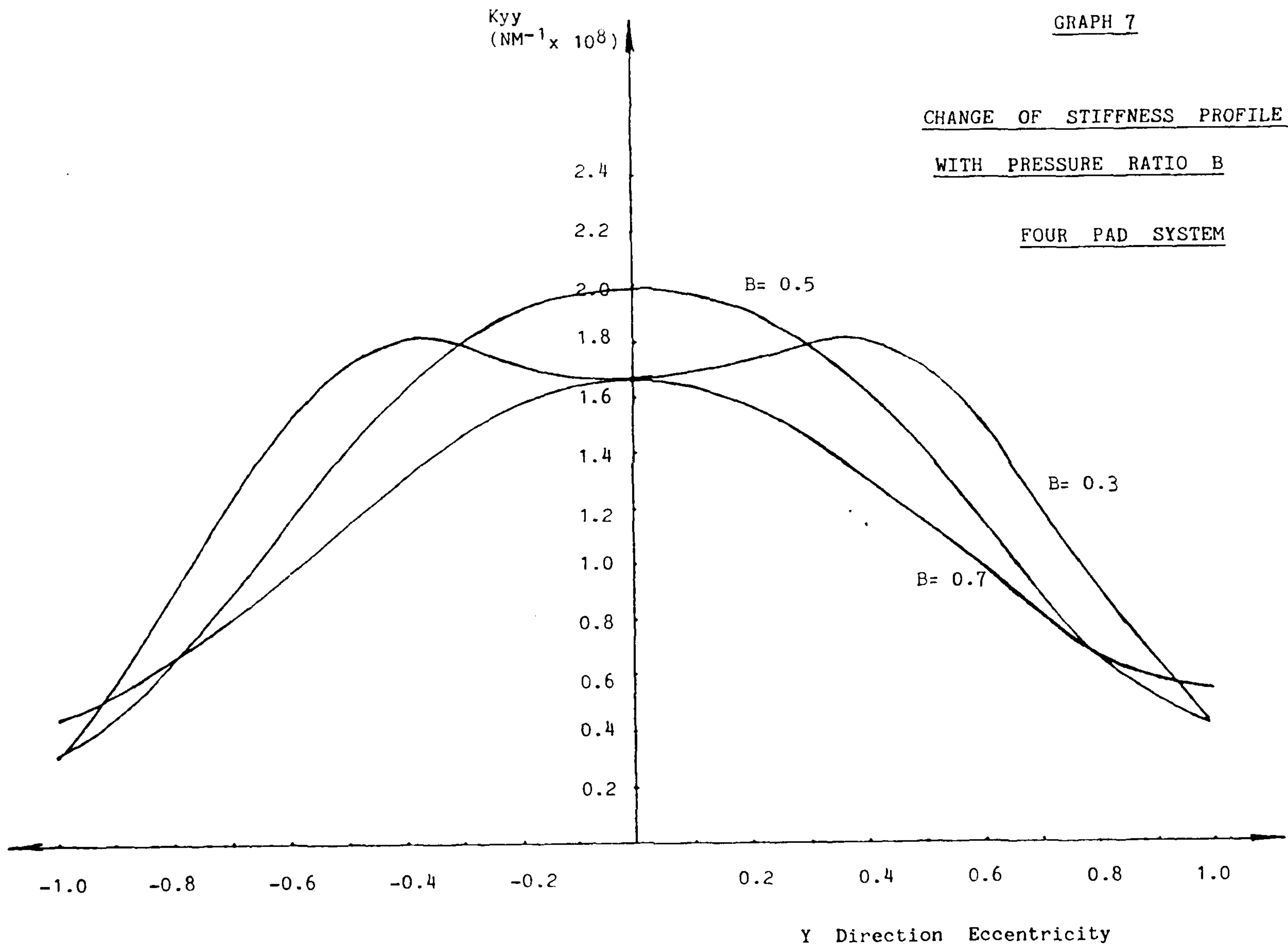
FOUR PAD SYSTEM

GRAPH 7

CHANGE OF STIFFNESS PROFILE Kyy

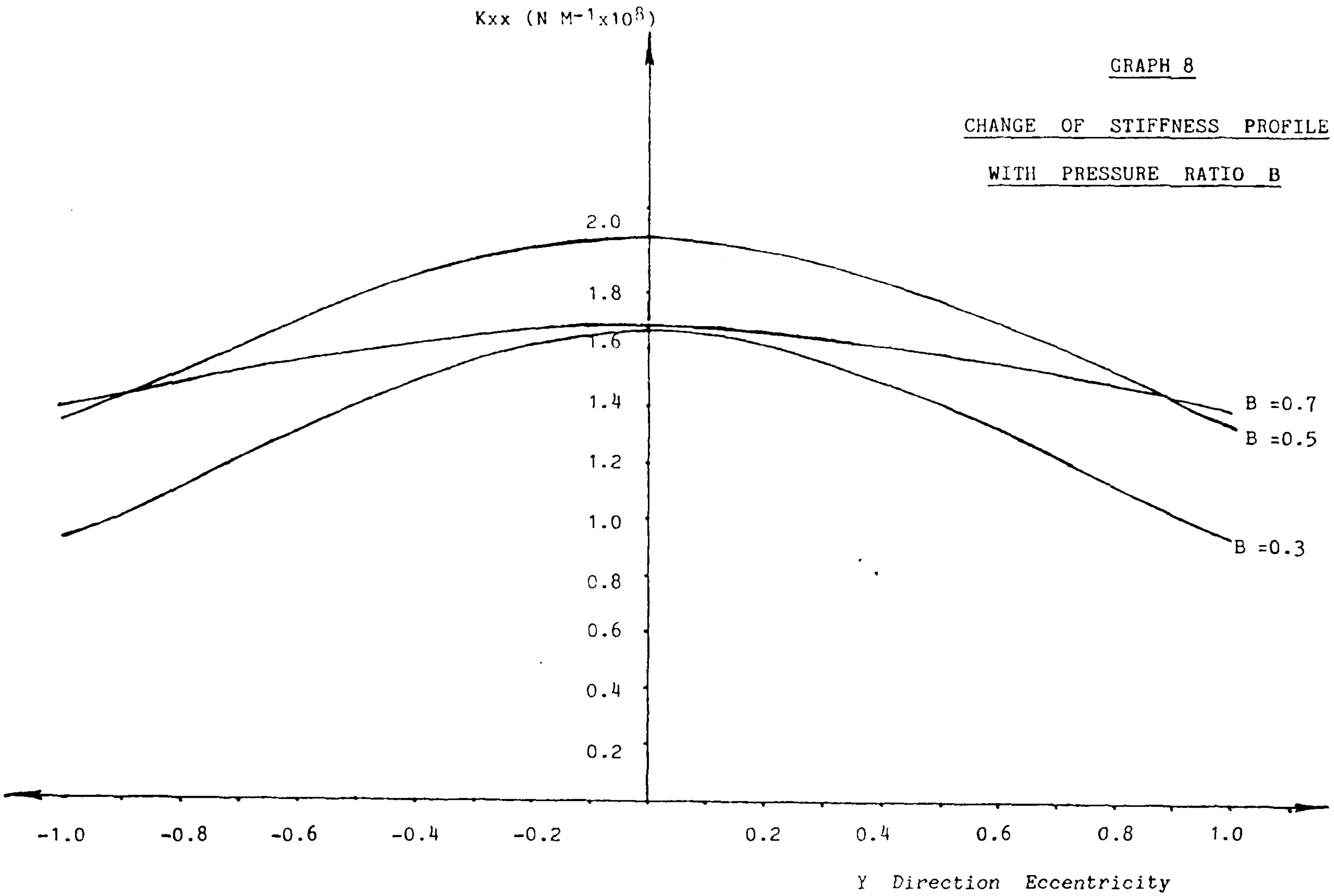
WITH PRESSURE RATIO B

FOUR PAD SYSTEM

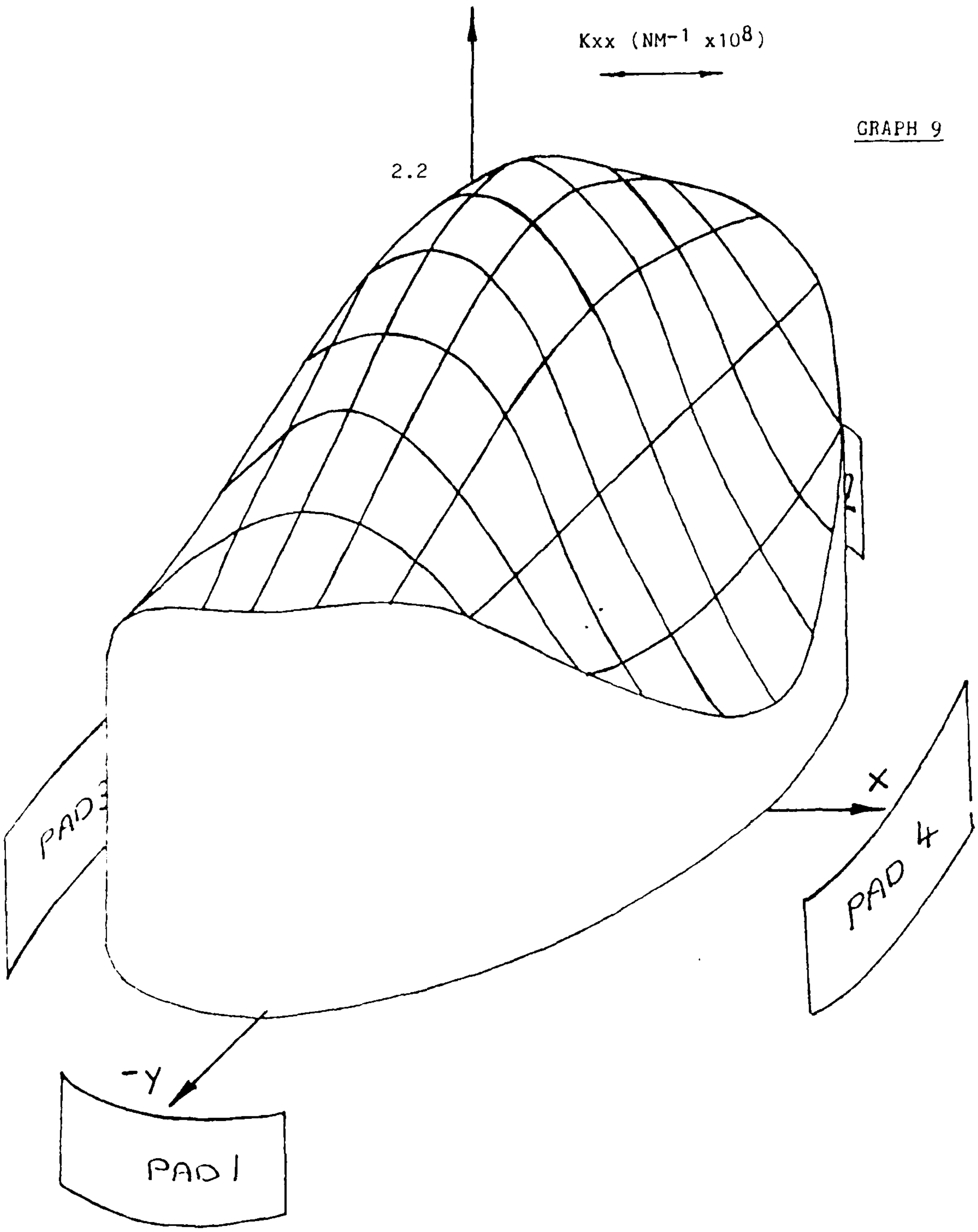


GRAPH 8

CHANGE OF STIFFNESS PROFILE WITH PRESSURE RATIO B



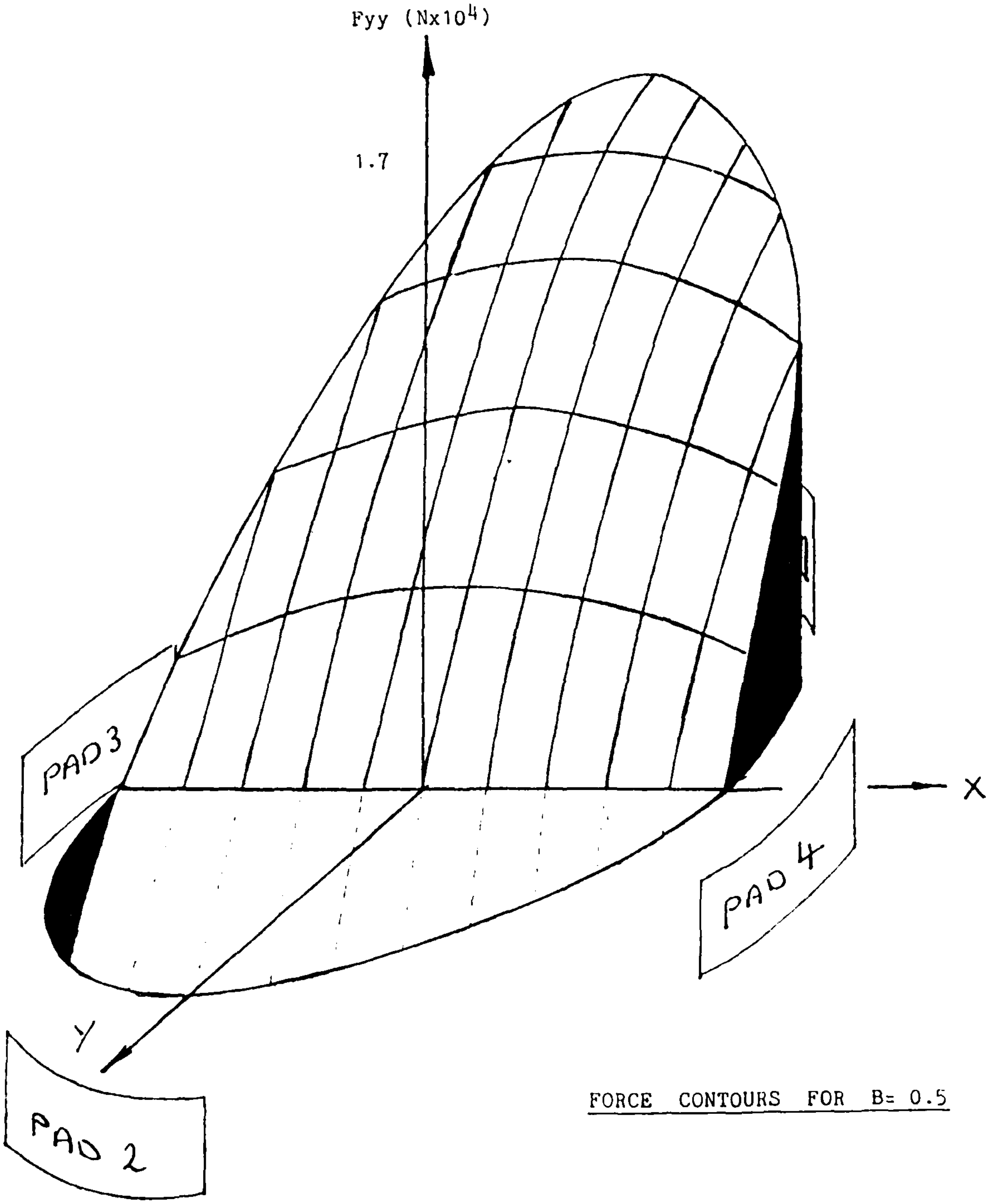
GRAPH 9



STATIC STIFFNESS PROFILE

WITHIN CLEARANCE CIRCLE

GRAPH 10

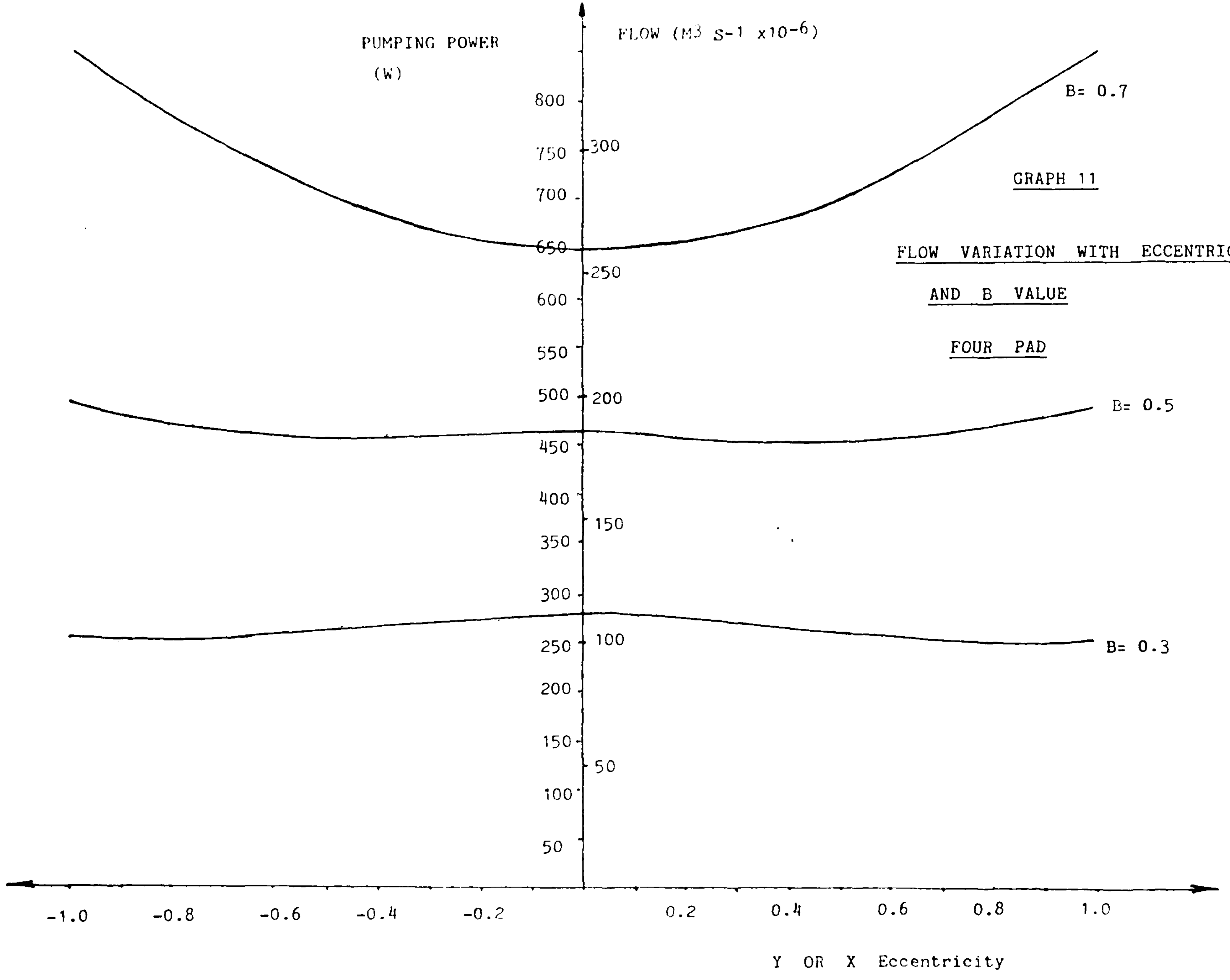


FORCE CONTOURS FOR B= 0.5

GRAPH 11

FLOW VARIATION WITH ECCENTRICITY AND B VALUE

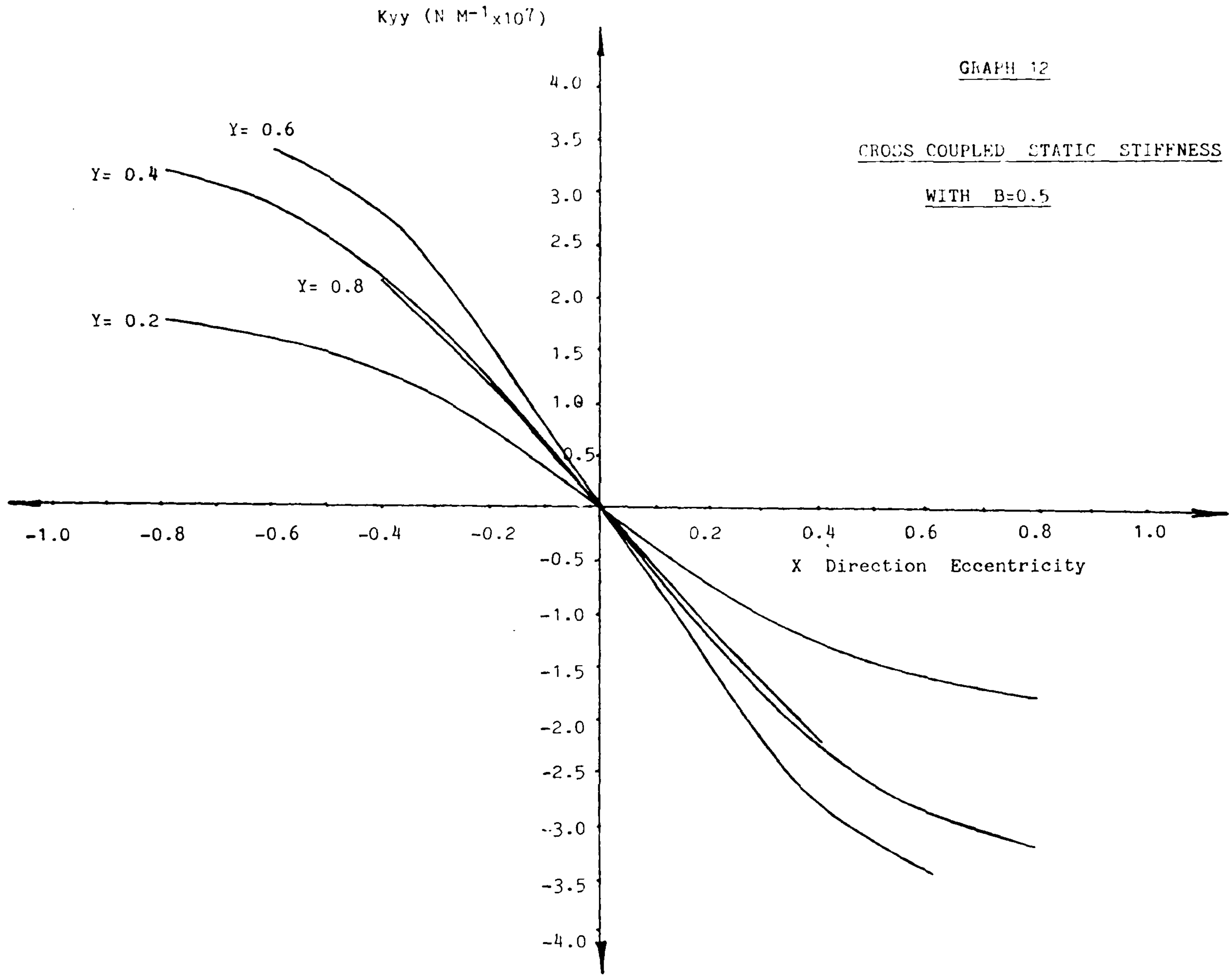
FOUR PAD



GRAPH 12

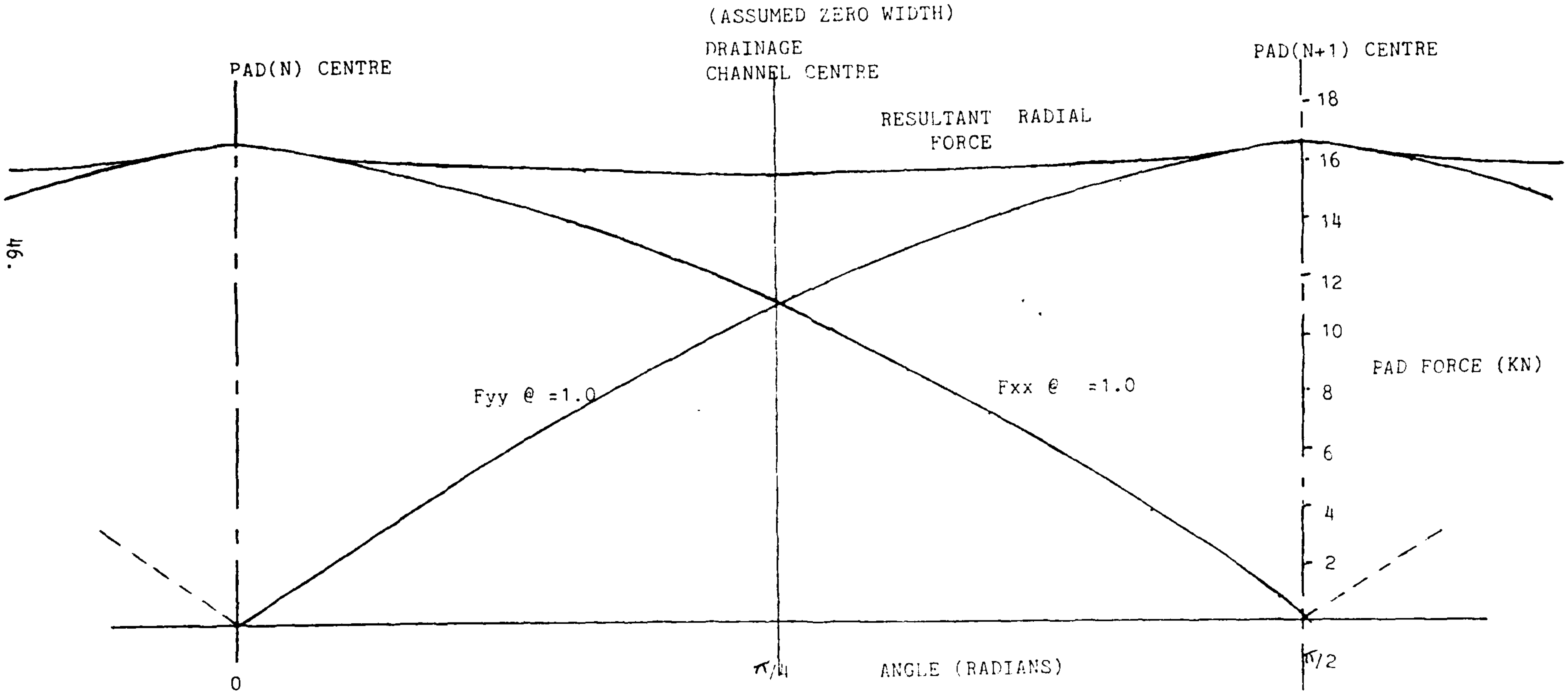
CROSS COUPLED STATIC STIFFNESS

WITH B=0.5



GRAPH 13

CHANGES IN RADIAL FORCE WITH ANGULAR VARIATION



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