MILITARY TECHNICAL COLLEGE

CAIRO - EGYPT

1- # SURFACE ROUGHNESS EFFECTS ON FLOW BETWEEN

2- # STATIONARY- AND ROTATING DISCS

3- A. I. EL-Batran*, M. EL-Kadi**, N. I. I. Hewedy***, B. A. Khalifa***

4- ## ABSTRACT

The turbulent outward fluid flow between stationary and a rotating disc enclosed by a cylindrical housing has been investigated experimentally . This type of flow occurs in many applications of engineering interests . Most of the flows encountered in practice are turbulent specially in turbo-machinery.

The effect of surface roughness on the frictional moment of the rotating disc as well as on the radial pressure distributions has been investigated for different relative roughness ratios of the rotating disc (R/k), $[600 \le (R/k) \le 1640]$. Here k is the mean particle diameter of the used grain size and R is the outer disc radius . The flow is investigated at a dimensionless gap height (H) of $[0.013 \le (H=h/R) \le 0.40]$, mass flow coefficient (C_w) of $[1.0\times10^3 \le (C_w = \rho Q/\mu R) \le 4.2\times10^3]$ and rotational Reynolds number (Re) of $[3\times10^5 \le (Re = \rho\omega R^2/\mu) \le 1.23\times10^6]$. Radial pressure and frictional moment for rough discs are compared with the experimental results obtained on smooth rotating discs . The results have showed that the surface roughness affects strongly upon the radial pressure distribution, the frictional moment and the axial thrust force .

INTRODUCTION

This paper concerns the effects of rotating disc roughness on the radial pressure and frictional moment distributions for flow in axial gap between a stationary and a rotating disc, enclosed in a cylindrical housing . The results of smooth rotating disc have been presented in [5] . The effect of surface roughness have been investigated for enclosed rotating discs with out through flow or free discs in many works . So some works are reviewed here . The effect of surface roughness on the frictional resistance of enclosed rotating discs have have been studied experimentally by [1] . Torque were obtained
5- over the range of rotational Reynolds numbers Re, 4×10^3 to 6×10^6

* Graduate student, ** Lecturer, *** Professor, Department . of Mechanical Power Engineering, Faculty of Engineering & Tech , Menoufia University, Shebin El-Kom, Egypt .

, three different relative roughness R/k of 1000, 2000 and 3200 and three axial gap ratio h/R of 0.0227, 0.0609 and 0.112 . A comparison between smooth and rough discs were obtained also . The effect of surface roughness on the frictional moment of a rotating conical discs were investigated by [2] experimentally. Torque was measured for six vertex angle from 30^o to 180^o, rotational Reynolds number ranged between $2x10^5$ to $2x10^6$ and relative roughness from 1000 to 3200 for enclosed system and with out through-flow .

Bayley and Conway studied experimentally the flow in a narrow gap between rotating and a stationary disc in [3] . The torque and pressure were measured for a rotational Reynolds numbers from 0 to 10^7, mass flow coefficient from 0 to 10^4 for gap ratio (h/R) from 0.004 to 0.06 .

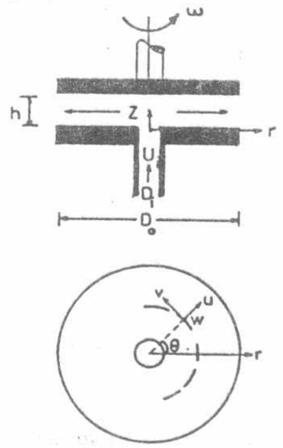


Fig.(1)

Schematic diagram for flow between stationary and rough rotating disc .

In this work the effect of four different surface roughness has been investigated experimentally on the radial pressure distributions, axial thrust coefficients C_{th} and the frictional moment coefficients C_m has been tested in details for different axial gaps, (h) different rotational Reynolds numbers (Re) and different mass flow coefficients (C_w) .

The problem to be measured is sketched in Fig.(1); (h) is the gap height between the discs . The fluid enters the domain axially through the circular opening of diameter (D_i) in the stationary disc and then fluid flow radially out-ward toward the rotating disc ..

EXPERIMENTAL APPARATUS

A testing apparatus was designed to carry out the experimental work of this investigation . The apparatus is shown schematically in Fig.2 . Water was used as a test fluid in the experimental work which flows through the inlet opening (20) in the stationary disc of 10 mm diameter and fills the system of the cylindrical housing which contains the discs, see Fig.(2) .

(1) Aluminum head
(2) 3-Micrometers
(3) Front plate
(4) O - ring
(5) Bourdon-tube gauge
(6) Cylindrical housing
(7) Back plate
(8) 8-Fixing bolts
(9) Stuffing box
(10) 2-Bearings
(11) Fixing bush
(12) Coupling
(13) DC-motor
(14) Stationary disc
(15) Rotating disc
(16) 12-Pressure taps
(17) 3-Adjusting bolts
(18) Outlet opening (19) Rotating shaft (20) Inlet opening .

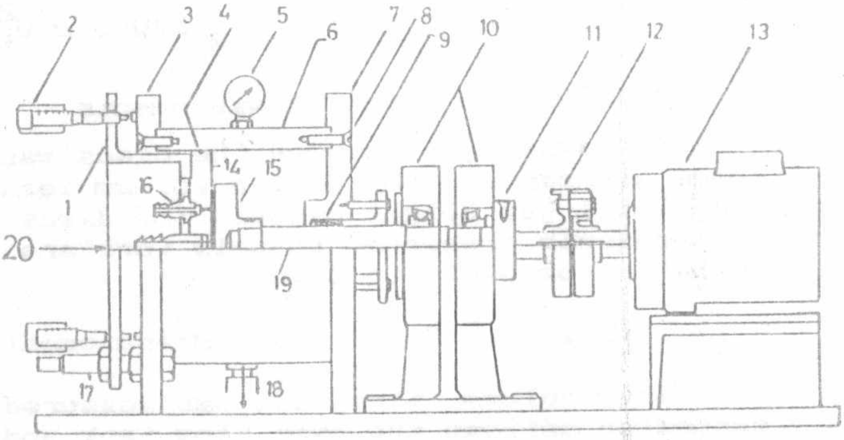


Fig.2. Schematic layout of apparatus

The rotating disc (15) is the same diameter as that used in the previous study [5] with 150 mm outside diameters and 15 mm thickness but it has two circular grooves with 25 and 60 mm radius, 5 mm width and 2 mm depth are formed on it's surface. A DC-motor (13) was coupled to the shaft by the coupling (12) . A special electrical circuit provided with a variable voltage regulator [Auto-Transformer] was arranged to produce a variable output voltage which produces variable motor speeds .The supply circuit is consisted mainly of a centrifugal water pump , main upper tank and an overhead tank to get a constant static supply head during the test . More details of the experimental apparatus could be obtained in [5] .

From the dimensional analysis it could be easily shown that the pressure distribution and the frictional moment depend upon five principle independent variables , namely ; the dimensionless gap height (H), the rotational Reynolds number (Re), the mass flow coefficient (C_w), relative roughness (R/k) and the radius ratio $\bar{R}$.

MEASURING METHODS

Pressure Measurements

Radial pressure distributions on the stationary flat disc were obtained from measurements obtained from the static pressure holes along the radius by using a bank of U-tube manometers with Carbon Tetrachloride (CCl_4) as a liquid of measuring . The pressure at each radius was measured as a head difference and written non-dimensionally as the pressure coefficient C_p , where

$$C_p = [(P_r - P_R)/0.5\,\rho\,U_i^2] \qquad (1)$$

Axial Gap Height Measurements

The axial gap between the discs was measured by using the three micrometers (2) having .01 mm least-count step, see Fig.2. The axial gap height between the discs could be varied and the measurements presented in this work are in the ranges of H, (H=h/R) from 0.013 to 0.40 .

Mass Flow Coefficient Measurements (C_w)

The volume flow rate was measured by a venturi-meter connected between the over flow tank and inlet flow pipe . The venturi-meter is calibrated by collecting tank and the carbon tetrachloride is used also to get the pressure drop measurements between the inlet and the throat points . The inlet and throat area ratio of venturi is designed as 0.5 . The volume flow rate of the working fluid was controlled by the outlet-opening (18) , Fig.2 . From these data the mass flow coefficient (C_w) was determined as :

$$C_w = \rho\,Q/\mu\,R \qquad (2)$$

Rotational Reynolds Number Measurements (Re)

The rotational speeds which are required for determining (Re) were measured by a digital speedo-meter which gives the number of revolutions per minute (N) directly .The angular velocity (ω) and the Reynolds number (Re) could be obtained as:

$$\omega = (2\,\pi\,N/60) \quad \& \quad Re = (\rho\,\omega\,R^2)/\mu) \qquad (3)$$

where ω and R are the angular velocity and the outer radius of the rotating disc respectively .

Frictional Moment Coefficient Measurements (C_m)

Frictional moments on the rotating disc were measured by an electrical digital Watt-meter which gives the power required to overcome the friction between the fluid and the disc . The measuring method was carried out as follows :

The power was measured which the housing was empty of water at the required speeds (no load power), P_e . Then the power was measured which the water filling the housing in the normal running operation (running load power), P_l . After that the frictional power can be calculated as follows :

Frictional power = running load power - no load power

$$P = P_l - P_e \quad \text{where,}$$

$$P = m \times \omega \quad \& \quad C_m = m/0.5\rho\omega^2 R^5 \qquad (4)$$

where ω , m and R are the angular speed, frictional moment and the outer radius of the rotating disc respectively .

Axial Thrust Coefficient (C_{th})

Axial thrust force is calculated numerically by the complete integration of the measured pressure distributions on the surface area of the stationary disc and is written as follows :

$$C_{th} = \int_{R_1}^{R} [(P_r - P_R)\, 2\pi r dr] / [\rho \pi \omega^2 R^2 (R^2 - R_1^2)] \tag{5}$$

Surface Roughness

Surface roughness for these rough-disc tests were obtained by cementing commercial waterproof grit papers to the surface of the rotating disc . A poison kit cement was used as the adhesive,and gasket cement applied in a fillet around the edges of the paper served as a sealer . The grit papers used with standard mesh number and mean grit diameter, (k) and written non-dimensionally as relative roughness as (R/k) . The surface roughness used in the present work is shown in table. I

Table.I

Grit size	k, mm.	R/k
#120	0.250	600
#180	0.083	900
#280	0.053	1415
#320	0.046	1630

EXPERIMENTAL RESULTS AND DISCUSSION

This section is concerned with the results of measurements and the effect of measured parameters ; Re, C_w, H and surface roughness on radial pressure, frictional moment and the axial thrust force distributions .

Radial Pressure Distributions

The effects of changing the relative surface roughness on radial pressure distribution are plotted in Fig.(3) . The smooth disc pressure distributions are also included in the same curves for comparison . Fig.3(a) shows the radial pressure profiles at constant rotational Reynolds number, mass flow coefficient, small gap height, H=0.013 and four different relative roughness .In addition, the effect of changing the gap height, mass flow coefficient and the relative roughness upon the radial pressure distribution are shown in Fig.3(b, c, d) . It is clear that increasing the surface roughness of the rotating disc increases the negative pressure coefficient amplitude . Also, the effect of relative roughness on the radial pressure distribution is reduced with increasing the gap height . The same results are obtained with increasing the mass flow coefficient . The effect of changing the gap height on the radial pressure distribution for constant relative roughness, mass flow coefficient and rotational Reynolds number is shown in Fig.3(e) . It is shown that increasing the gap height decreases the negative pressure coefficient and this decrease is small in the bigger gap heights . The effect of varying the rotational rotational Reynolds number on radial pressure distribution is shown in Fig.3(f) . Increasing the rotational Reynolds number increased the pressure coefficient amplitude . In all curves the pressure drop increase at the inlet region due to the shape of the inlet flow region and the small inertia due to rotation. In all profiles the effect of roughness is very clear .

The obtained pressure distribution is integrated to give the axial thrust coefficient, Eqn.(5) . Its variation against the mass flow coefficient is shown in Fig.(4) at variable values of surface roughness and constant rotational Reynolds number and gap height, H=0.267 . The curves show that surface roughness of the rotating disc has increased the negative value of the axial thrust coefficient . Also the curves show that increasing the mass flow coefficient decreases the axial thrust coefficient amplitude .

Disc Frictional Moment Distributions

The Effect of surface roughness on frictional moment are represented in Fig.(5) . Fig.5(a) show the effect varying the surface roughness of the rotating disc on frictional moment for constant mass flow coefficient and gap , H=0.027 . The effect of surface roughness on frictional moment for big gap height is shown in Fig.5(b) . It is shown that increasing the surface roughness increases the frictional moment on the rotating disc. Also, the big gap decreased the frictional moment .A comparison between the results of the rough and smooth discs is plotted and the different is clear at the curves . Generally increasing the rotational Reynolds number decreases the frictional moment on the rotating disc .

Comparison Between Measurements and Theories

For the experimental results to be credible a comparison between the measured pressure at different rotational Reynolds number and the theory of [7] is shown in Fig.(6) . The solid lines represent the theoretical results for the smooth disc . The dashed line with symbols represent the rough disc at same rotational Reynolds number , $Re=1.1x10^6$. It is clear that a good agreement between the experimental results and the theory for the smooth disc, the difference at rough case due to the surface roughness which was not taken in the consideration in the theoretical solution . Another comparison is presented in Fig.(7) between the present frictional moment and results of [2] for the case of smooth and rough discs at the same gap height, mass flow coefficient and surface roughness . It is clear that the agreement is good in the range of measurements.

CONCLUSIONS

The principle conclusions to be drawn from the present experimental work are as follows :

1. Increasing the surface roughness of the rotating disc increases the radial pressure amplitude coefficient, therefore the axial thrust is increased .

2. The effect of relative roughness of the rotating discs upon the radial pressure distribution is reduced with increasing the gap height . While the effect is more pronounced with increasing the rotational Reynolds number .

4. Increasing the surface roughness of the rotating disc has increased the frictional moment .

5. The comparison between the present work and the previous works has shown the agreement is quite satisfactory .

REFERENCES

[1] DAILY, J.W. and NECE, R.E. '' Chamber Dimension effects on Induced Flow and Frictional Resistance of Enclosed Rotating disks'' Journal OF BASIC ENGG, MAR,(1960), pp.217-232.

[2] DAILY, J.W. and NECE, R.E. ''Roughness Effects On Frictional Resistance of Enclosed Rotating Disks '' Journal of Basic ENGG, SEPT,(1960), pp.217-232.

[3] BAYLEY, F.J. AND OWEN, J.M. ''Flow Between a Rotating and a Stationary Disc .'' JOURNAL OF AERONAUTICAL QUARTERLY , JULY ,(1968) ,PP.129-147 .

[4] YAMADA, Y. AND ITO, M.'' On Frictional Resistance Of Enclosed Rotating Cones '', (2nd Report, Effect of Surface Roughness), Bulletin of the JSME, Vol.19, No.134, August, (1976), pp. 943-950.

[5] El-BATRAN, A .I . ''Fluid Flow Between a Stationary and a Rotating Disc '', M.sc.Thesis Faculty Of Engineering & Technology, Shebin El Kom Egypt , (1988) .

[6] KUROKAWA, J. and SAKUMA, M. '' Flow in a Narrow Gap Along an Enclosed Rotating Disk With Through -Flow.'', JSME INTERNATIONAL JOURNAL, Vol.31, No.2, (1988), pp. 243-250.

[7] KHALIFA,B.A., HEWEDY, N.I.I., EL-KADI, M. and EL-BATRAN, A.I., '' Numerical Computations of Flow Between a Stationary and a Rotating Disc With Two-Equations Model of Turbulence '', Paper in the Seventh International Conference for Mechanical Power Engineering, 17-20 DECE, (1990), Cairo University, Egypt .

NOMENCLATURE

C_p	pressure coefficient , Eqn.(1)
C_w	mass flow coefficient , Eqn.(2)
C_m	frictional moment coefficient , Eqn.(4)
C_{th}	axial thrust coefficient , Eqn.(5)
D_i	diameter of the inlet opening = $2R_i$
D_o	outer diameter of the rotating disc = $2R$
H	dimensionless inlet axial gap height = (h/R)
h	gap height between discs at inlet
k	mean grit particle diameter
m	frictional moment on the rotating disc
P_r, P_R	pressure at any radius and outlet radius respectively
P	power of the electric motor , Eqn.(4)
Q	volume flow rate
Re	rotational Reynolds number , Eqn.(3)
$\bar{R}$	dimensionless radius ratio = (r/R_i)
r, θ, z	radial -, angular - , & axial coordinates, Fig.1
u, v, w	radial-, tangential-, &axial velocity components, Fig.1
U_i	average velocity of the incoming fluid
ρ, μ	density and dynamic viscosity of the fluid
ω	angular velocity of rotating disc

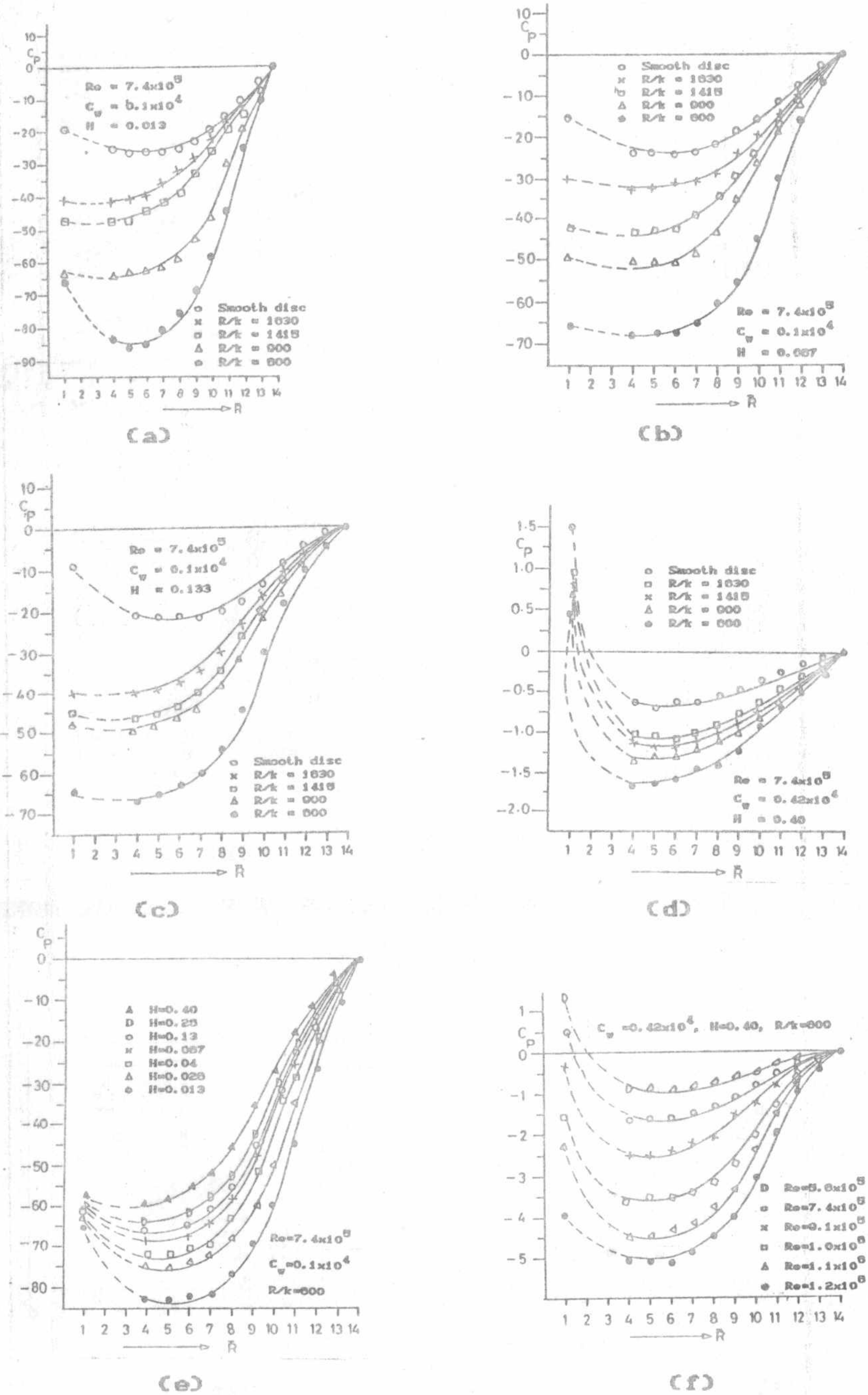


Fig. 3 EFFECT OF SURFACE ROUGHNESS ON RADIAL PRESSURE

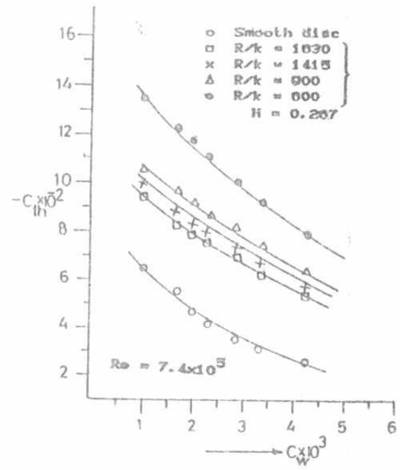
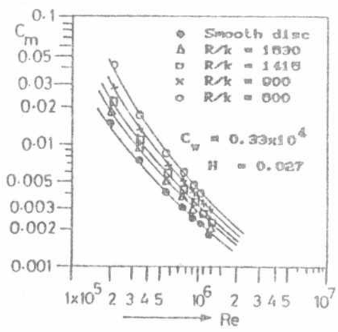
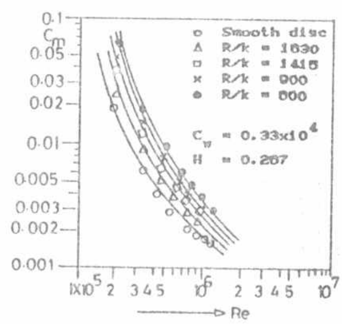


Fig. 4

EFFECT OF SURFACE ROUGHNESS ON AXIAL THRUST COEFFICIENT



(a)



(b)

Fig. 5 EFFECT OF SURFACE ROUGHNESS ON FRICTIONAL MOMENT

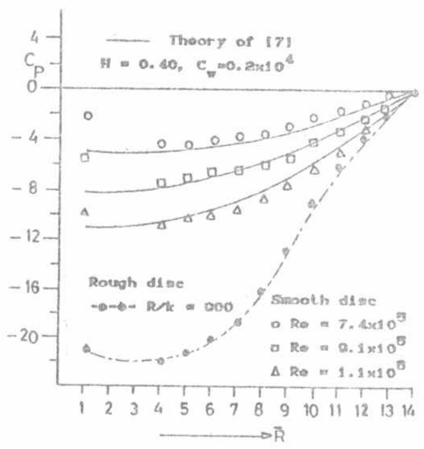


Fig. 6
COMPARISON BETWEEN PRESENT AND OTHER PRESSURE RESULTS

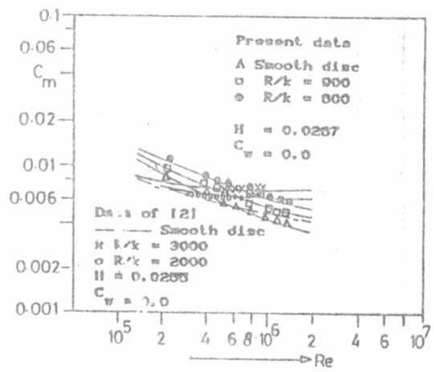


Fig. 7
COMPARISON BETWEEN PRESENT AND OTHER FRICTIONAL MOMENT