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DYREMI—Computer Software for Dynamics of Reciprocating Machine Installations*

J. S. RAO
HARMIT SINGH

Department of Mechanical Engineering, Indian Institute of Technology, New Delhi 110016, *India*

DYREMI (Dynamics of Reciprocating Machine Installations) is a software developed for the purpose of computer-aided instruction and learning during a first course in mechanism and machine theory. The program determines the residual inertia force of each cylinder, bearing loads and crankshaft torque, and displays the results in graphical as well as tabular form. For a multi-cylinder engine, the crankshaft torque and the desired flywheel size based on a proposed fluctuation in speed or the permissible maximum acceleration on sudden removal of load, are also determined. The program further determines the balancing requirements for the rotating masses in two planes and outputs the primary and secondary unbalance conditions of the engine.

AUTHORS' QUESTIONNAIRE

1. The paper discusses material/software for a course in

   Kinematics of Machinery (L-T-P) 2-1-0 (third semester Bachelor's degree); Dynamics of Machinery 3-1-2 (fourth semester Bachelor's degree).

2. Departments:

   Mechanical Engineering

3. Level of the Course (year):

   2nd year of four year program.

4. Mode of presentation:

   Classroom for lectures with PCs for special tutorial classes.

5. Is the material presented in a regular or elective course:

   Regular

6. Class or hours required to cover the material:

   This package is introduced during the second half of the fourth semester after the student has been exposed to kinematics of reciprocating machines during the third semester, force analysis (combined static and inertia), turning moment diagrams, flywheel and balancing of rotating and reciprocating parts during the fourth semester. Two to three hours of recap and further explanation of the system followed by two hours of tutorials.

7. Student homework or revision hours required for the materials:

   4 hours.

8. Description of novel aspects presented in your paper:

   The computer package gives the student a clear understanding on the application of kinematics and kinetics of reciprocating machine installations and the practical aspects of matching a reciprocating engine with a load.

9. Standard text recommended in the course, in addition to author's notes:

   Mechanism and Machine Theory, J. S. Rao and R. V. Dukkipati along with a Solution Manual, John Wiley (1989) and Wiley Eastern 2nd edn (1992).

10. The material is covered in the text, but not the computer program.

1. INTRODUCTION

COMPUTERS are becoming common tools for instruction at the undergraduate level. Several courses are being developed for computer-aided instruction and learning. Mechanism and machine theory (theory of machines, kinematics of machinery and dynamics of machinery) is a subject that can be easily adopted in this respect.

In the subject of mechanism and machine theory,

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a significant amount of time is spent on reciprocating engine kinematics, static and inertia force analysis and understanding of fluctuations in the torque delivered by the engine. Multicylinder engine dynamics leading to flywheel design requirements and balancing conditions of rotating and reciprocating parts follows the single cylinder engine analysis. The student is generally exposed to graphical and/or analytical methods for solving the problems manually, for a clear understanding of the subject. However, these methods are time-consuming and, therefore, it is advisable to use a computer. At the end of instruction, the student (or a designer) can understand and appreciate the above subject matter, if he can look at the problem as a whole, starting from kinematics of a single cylinder and progressing to a multicylinder engine installation. With this purpose in mind, a program is developed so that the student can interact with a PC, in giving all the input required for the analysis and observe the results in graphical as well as tabular form.

2. THEORY

Consider a single-cylinder reciprocating engine, as shown in Fig. 1. The basic equations involved are briefly outlined below, see also Rao and Dukkipati [1].

2.1 *Acceleration of piston*

TMS is given by:

$$\alpha = -\omega^2 r\left[\cos\theta + \frac{(n^2-1)\cos 2\theta + \cos^4\theta}{n^3(1-\sin^2\theta/n^2)^{1.5}}\right] \quad (1)$$

where ω is the angular velocity in radians/s.

2.2 *Bearing loads*

The crankpin load, F_{32}, is

$$F^x_{32} = m_{3A}\omega^2 r\cos\theta - (m_{3B}+m_4)\,\alpha - p(\theta)A$$

$$F^y_{32} = m_{3A}\omega^2 r\sin\theta + \{(m_{3B}+m_4)\,\alpha - p(\theta)A\}\tan\phi \quad (2)$$

where:

m_{3A} is the mass of connecting rod located at the crankpin.
m_{3B} is the mass of connecting rod located at the wristpin;
$p(\theta)$ is the gas pressure, see Appendix;
A is the cross-sectional area of cylinder bore.

The main bearing load F_{12} is given by

$$F^x_{12} = p(\theta)A + (m_{3B}+m_4)\alpha - \omega^2 r\left(m_{3A} + m_w\frac{\overline{O_2G_2}}{r} + m_{cp} - m_{cb}\frac{\overline{O_2b_2}}{r}\right)\cos\theta$$

$$F^y_{12} = -\{p(\theta)A + (m_{3B}+m_4)\alpha\}\tan\phi - \omega^2 r\left(m_{3A} + m_w\frac{\overline{O_2G_2}}{r} + m_{cp} - m_{cb}\frac{\overline{O_2b_2}}{r}\right)\sin\theta \quad (3)$$

where

m_w is the crank mass (the two webs);
O_2G_2 is the distance of c.g. of the crank from the main bearing centre;
m_{cp} is the mass of the crankpin;
m_{cb} is the counterbalance mass;
O_2b_2 is the distance of c.g. of the counterbalance mass from the main bearing centre.

The wristpin load, F_{34}, is given by

$$F^x_{34} = m_4\alpha + p(\theta)A$$

$$F^y_{34} = -\{(m_{3B}+m_4)\alpha + p(\theta)A\}\tan\phi \quad (4)$$

where m_4 is the mass of piston and piston rings.

2.3 *Cylinder wall force and crankshaft torque*

The cylinder wall force, F_{41}, is given by

$$F^x_{41} = 0$$

$$F^y_{41} = -\{(m_{3B}+m_4)\alpha + p(\theta)A\}\tan\phi. \quad (5)$$

The crankshaft torque, T, is

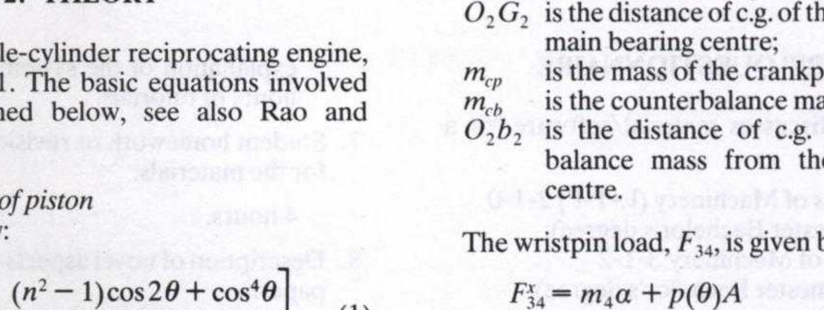

Fig. 1. Reciprocating engine analysis.

$$T = \{(m_{3B} + m_4)\alpha + p(\theta)A\}(\cos\theta + n\cos\theta)r\tan\phi. \quad (6)$$

The correction torque, T_c, to be aded to the above is

$$T_c = -m_3 b\omega^2(l - L)\frac{(n^2 - 1)\sin 2\theta}{2(n^2 - \sin^2\theta)^2} \quad (7)$$

where

b is the distance between the c.g. of connecting rod and crankpin;

$L = \tau^2 g/4\pi^2$ is the distance of centre of percussion of the connecting rod from the wristpin, with τ as the time period for one full swing of the connecting rod when suspended from the wristpin centre and

g is the acceleration due to gravity.

3. FLYWHEEL

In a multicylinder engine, it is necessary that all the cylinders fire in 720° (for 4-stroke engines) at equal intervals. The net torque delivered by the engine can be simply obtained by distributing individual cylinder torques in the 720° of the crank rotation and summing them up. From this diagram, the mean torque, T_m, and the maximum fluctuation of energy, E, are obtained.

A typical reciprocating engine installation is shown in Fig. 2. It generally consists of the engine, a damper, a coupling and the load, say a generator. The flywheel is usually connected to one half of the coupling on the engine side. To determine the flywheel size, it is necessary to calculate the total mass moment of intertia of the installation.

The mass moment of inertia of each cylinder is obtained from

$$I_c = m_w\overline{O_2G_2}^2 + m_b\overline{O_2b_2}^2 + (m_{cp} + m_{3A})r^2 + \tfrac{1}{2}m_j r_j^2\,\tfrac{1}{2}m_{cp}r_{cp}^2 + \tfrac{1}{2}(m_{3B} + m_4)r^2 \quad (8)$$

where subscripts j and cp refer to the journal and crankpin, respectively. The mass moments of inertia of the damper, coupling, generator, etc., are given in the input.

The program provides for the estimation of flywheel inertia to keep the fluctuation of speed within a specified limit, or to limit the maximum acceleration α, of the installation under sudden removal of load. In the first case, the net mass moment of inertia, I_{net} of the installation is determined from

$$I_{net} = \frac{E}{\delta_s\omega^2} \quad (9)$$

where δ_s is the coefficient of fluctuation of speed.

In the second case, I_{net} is determined from

$$I_{net} = \frac{T_m}{\alpha}. \quad (10)$$

The flywheel inertia is then obtained by subtracting the inertias of the engine, damper, generator, etc., of the installation from the larger value of I_{net} determined from equations (9) and (10) above.

4. BALANCING

A reciprocating engine is inherently unbalanced, as it consists of rotating and reciprocating parts as well as those describing plane motion, i.e. the connecting rod, in the system. For simplicity, the connecting rod is considered as rotating and reciprocating parts by lumping its mass at the crankpin and wristpins respectively.

4.1 *Shaking force*

To keep the shaking forces from each cylinder to a minimum, a portion of the reciprocating mass is balanced as a rotating part, which reduces the primary forces in the direction of stroke, however, this introduces an additional force in a direction

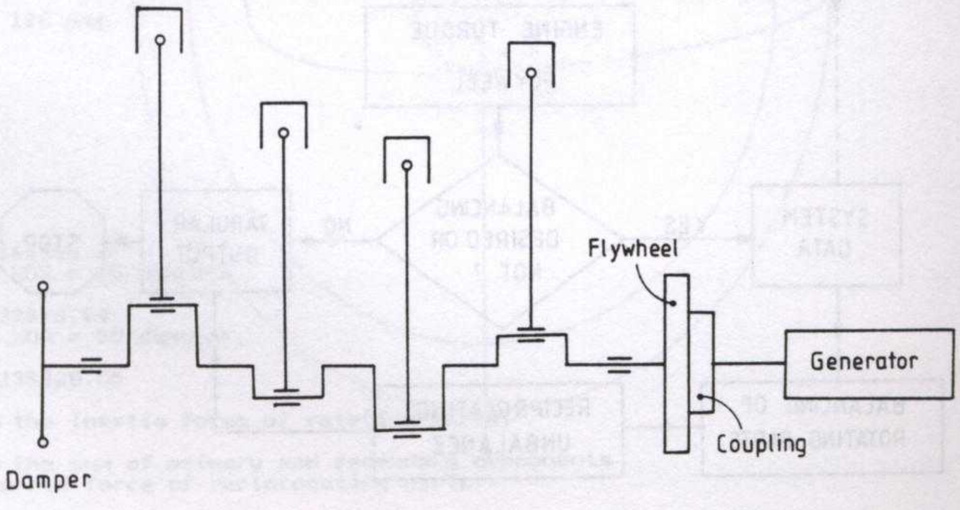

Fig. 2. Typical reciprocating machinery installation.

perpendicular to the stroke. The counterbalancing mass is chosen so as to keep the maximum value of this force within suitable values. The present program calculates the residual inertia force of each cylinder, considering the crank, counter-balancing masses and the reciprocating masses (including both the primary and secondary components) for one full revolution of the crank with the help of

$$F_v = m_{rot}\omega^2 \sin\theta$$

$$F_h = (m_{rot} + m_{rec})\omega^2 \cos\theta + \frac{1}{n} m_{rec}\omega^2 \cos 2\theta \quad (11)$$

where

$$m_{rot} = m_{cp} + m_w \frac{\overline{O_2G_2}}{r} + m_{3A} - m_b \frac{\overline{O_2b_2}}{r}$$

$$m_{rec} = m_{3B} + m_4.$$

4.2 *Balancing of rotating parts*

The rotating parts of all the cylinders in the engine (cranks arranged according to the firing angle), can be balanced by two rotating masses located in any two convenient planes. Normally the two planes chosen are the damper and the flywheel (or the coupling half connected to the engine). Let the damper be located at plane 0, the cylinders in 1, 2, ... N planes, and the flywheel in plane $(N+1)$.

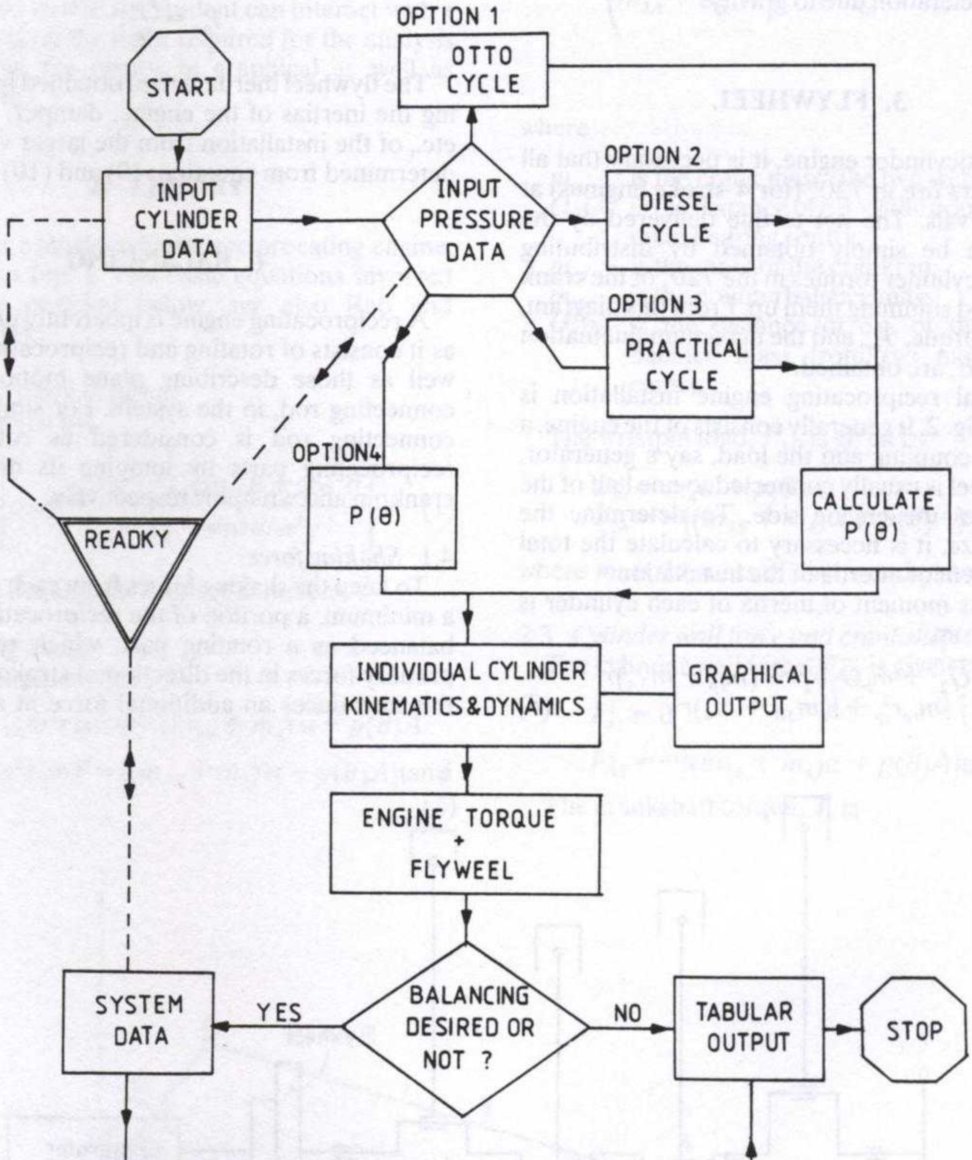

Fig. 3. Structure of computer program (DYREMI.PAS).

The mid plane of the engine is denoted as $(N+2)$. Let Φ_i be the angle made by the ith crank in the direction of rotation of the crankshaft from the reference crank (usually the first, i.e. $\Phi_i = 0$) as determined from the firing order of the engine and let d_{Li} be the distance of the ith cylinder from the damper plane, then the components of the sum of the moments of the unbalance forces in x and y directions are given by

$$M_{dx} = \sum_{i=1}^{N} m_{rot}\omega^2 r d_{Li} \cos\phi_i$$
$$M_{dy} = \sum_{i=1}^{N} m_{rot}\omega^2 r d_{Li} \sin\phi_i \qquad (12)$$

The balancing mass in kg.mm and its location in the flywheel plane can be obtained by making its moment components about the damper plane equal and opposite to the above. This procedure is repeated by taking moments about the flywheel plane and determining the balancing mass required in the damper plane along with its angular location.

4.3 *Primary and secondary unbalanced forces and moments due to reciprocating parts*

The primary unbalanced force of the engine reciprocating parts is

$$F_p = \sum_{i=1}^{N} m_{rec}\omega^2 r \cos(\theta_1 + \phi_i) \qquad (13)$$

where θ_1 is the angle made by the first crank from the line of stroke.

The secondary unbalanced force of the engine reciprocating parts is

$$F_s = \sum_{i=1}^{N} \frac{1}{n} m_{rec}\omega^2 r \cos 2(\theta_1 + \phi_i). \qquad (14)$$

The primary unbalanced moment of the engine reciprocating parts about a reference plane is

$$M_p = \sum_{i=1}^{N} m_{rec}\omega^2 r d_i \cos(\theta_1 + \phi_i). \qquad (15)$$

The secondary unbalanced moment of the engine reciprocating parts about a reference plane is

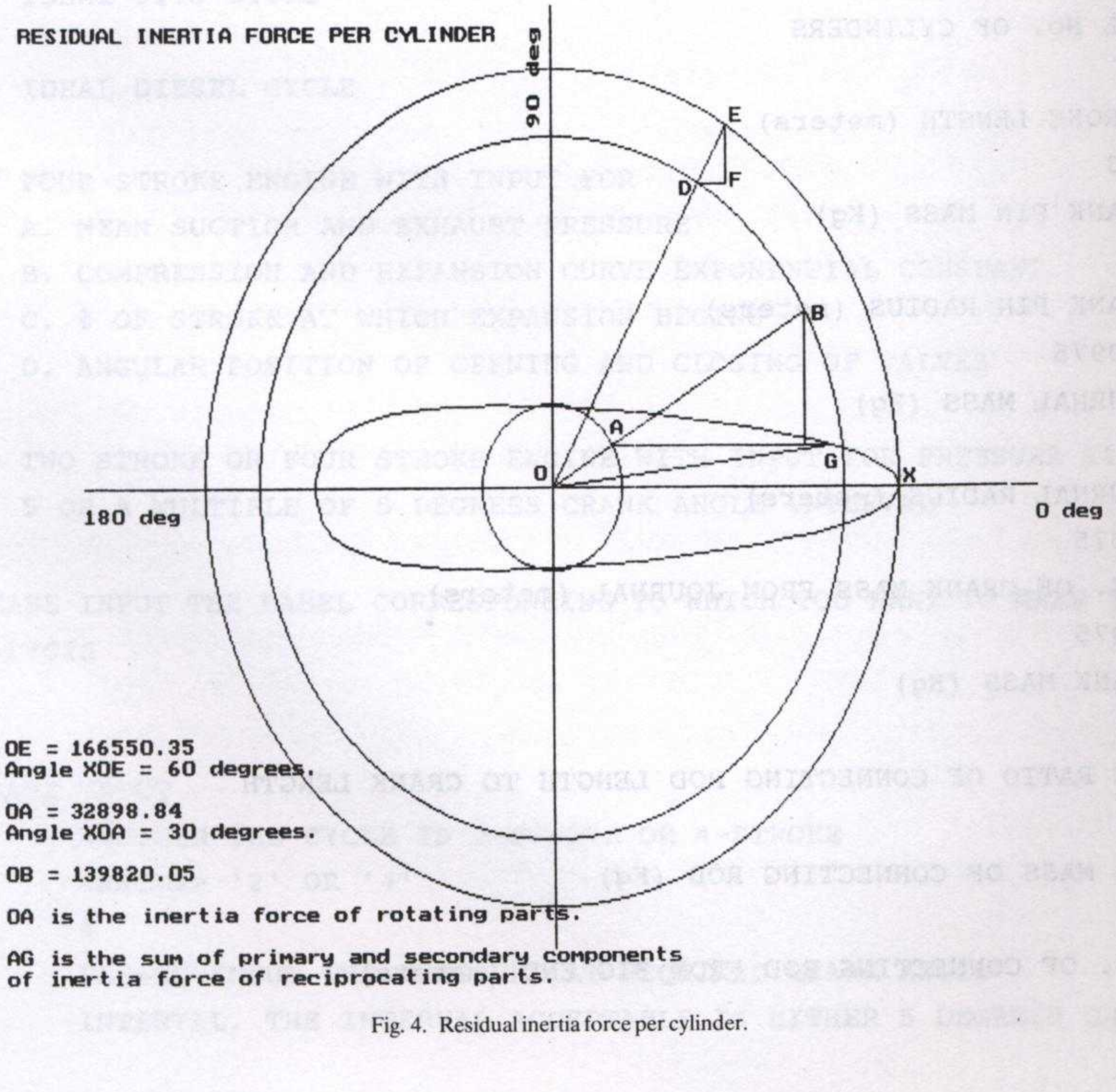

Fig. 4. Residual inertia force per cylinder.

$$M_s = \sum_{i=1}^{N} \frac{1}{n} m_{rec}\omega^2 d_i r \cos 2(\theta_1 + \phi_i). \quad (16)$$

5. COMPUTER PROGRAM

The program is developed in Borland Turbo Pascal Version 5.5 with .CHR and .BGI and needs MS DOS version 4.01 with GRAPHIC.COM and GRAPHIC.PRO. The structure of the program is given in Fig. 3, which is self-explanatory. The main program is DYREMI.EXE, which incorporates the above analysis. It uses READKY.PAS for data input in a unique manner, as explained in the next section.

6. INPUT TO PC

Make a directory DYREMI on the hard disk and load the required files including DYREMI.EXE. DYREMI executes the program and asks for the input in an interactive mode, which is to be given in response to the questions from the monitor. They are given here in sequence as they appear on the monitor while running the program. (Note: (1) in decimal quantities below, a digit should precede and follow the decimal; (2) use of the backspace key removes the current quantity being entered.)

The data given here belongs to a typical reciprocating engine driven generator set.

```
Cylinder Data

PLEASE INPUT
THE No. OF CYLINDERS
9
STROKE LENGTH (meters)
0.3
CRANK PIN MASS (Kg)
25
CRANK PIN RADIUS (meters)
0.0975
JOURNAL MASS (Kg)
30
JOURNAL RADIUS (meters)
0.075
C.G. OF CRANK MASS FROM JOURNAL (meters)
0.075
CRANK MASS (Kg)
90
THE RATIO OF CONNECTING ROD LENGTH TO CRANK LENGTH
4
THE MASS OF CONNECTING ROD (Kg)
60
C.G. OF CONNECTING ROD FROM BIG END (meters)
0.2
```

```
TIME FOR 60 OSCILLATIONS (secs)
93
THE RADIUS OF CYLINDER (meters)
0.15
MASS OF PISTON + PINS (Kg)
45
WRIST PIN RADIUS (meters)
0.05
COUNTER BALANCING MASS (Kg) & ITS C.G. (meters)
90
0.15
```

Pressure Data

```
THE PROGRAM ALLOWS FOR THE ANALYSIS OF
1 - IDEAL OTTO CYCLE

2 - IDEAL DIESEL CYCLE

3 - FOUR STROKE ENGINE WITH INPUT FOR
   A. MEAN SUCTION AND EXHAUST PRESSURE
   B. COMPRESSION AND EXPANSION CURVE EXPONENTIAL CONSTANT
   C. % OF STROKE AT WHICH EXPANSION BEGINS
   D. ANGULAR POSITION OF OPENING AND CLOSING OF VALVES

4 - TWO STROKE OR FOUR STROKE ENGINE WITH INPUT FOR PRESSURE AT
   5 OR A MULTIPLE OF 5 DEGREES CRANK ANGLE INTERVAL

PLEASE INPUT THE LABEL CORRESPONDING TO WHICH YOU WANT TO MAKE THE
ANALYSIS
4

PLEASE INPUT
     WHETHER THE CYCLE IS 2-STROKE OR 4-STROKE
     <ENTER> '2' OR '4'
     4
     PLEASE INPUT THE PRESSURE AT REQUIRED CRANK ANGLE
     INTERVAL. THE INTERVAL ACCEPTABLE IS EITHER 5 DEGREES OR
```

```
    ITS MULTIPLE. IF THE INTERVAL IS A MULTIPLE OF 5 DEGREES
    THEN LINEAR INTERPOLATION OF PRESSURE IS ASSUMED BETWEEN
    THE INPUT VALUES

    CRANK ANGLE                     PRESSURE
    (degrees)                       (Pascals)
       0                            0
       5                            68560
       90                           1234080
       180                          0
       720                          0
```

Engine and System Data

```
PLEASE INPUT
   THE ROTATIONAL SPEED (rpm)
   1000
   DAMPER INERTIA (Kg*sqr(meter)).
   20
   ADDITIONAL INERTIA (Kg*sqr(meter)) e.g. GENERATOR, COUPLING
   OTHER THAN ENGINE, DAMPER. IF ADDITIONAL INERTIA IS ABSENT
   ENTER THE VALUE AS 0.
   THE FLYWHEEL TO BE CALCULATED WILL BE ASSUMED TO BE AT THE
   POSITION OF THE COUPLING.
   200
   COEFFICIENT OF FLUCTUATION OF SPEED AS PERCENTAGE
   0.1
   MAX. ACCELERATION IF LOAD IS TAKEN OF SUDDENLY AS % RPM/S
   25
WOULD YOU LIKE TO FIND THE UNBALANCE PRIMARY/SECONDARY FORCES,
COUPLES IN DIFFERENT PLANES. IF YES ENTER 'Y' OR 'y' ELSE 'N' OR
'n'.
Y
PLEASE INPUT
     THE DISTANCE BETWEEN CYLINDER 1 AND 2
     0.4
     THE DISTANCE BETWEEN CYLINDER 2 AND 3
     0.4
```

```
          THE DISTANCE BETWEEN CYLINDER 3 AND 4
          0.4
          THE DISTANCE BETWEEN CYLINDER 4 AND 5
          0.4
          THE DISTANCE BETWEEN CYLINDER 5 AND 6
          0.4
          THE DISTANCE BETWEEN CYLINDER 6 AND 7
          0.4
          THE DISTANCE BETWEEN CYLINDER 7 AND 8
          0.4
          THE DISTANCE BETWEEN CYLINDER 8 AND 9
          0.4
          FOR DAMPER AND FLYWHEEL/COUPLING PLEASE INPUT THE
          DISTANCE (meter) WRT CYLINDER 1.IF PRECEDING 1
          DISTANCEBE ENTERED -VE ELSE +VE
          -0.5
          3.8
PLEASE INPUT THE FIRING ORDER
1 7 4 2 8 6 3 9 5
PLEASE INPUT
          THE PLANE No. ABOUT WHICH THE UNBALANCED COUPLE DUE TO
          ROTATING MASSES IS TO BE FOUND
          FOR DAMPER INPUT PLANE No.0
          FOR FLYWHEEL INPUT PLANE No. 10
          FOR MIDPLANE INPUT PLANE No.11
          11
THE COUPLE WILL BE BALANCED IN THE DAMPER AND FLYWHEEL/COUPLING
PLANE
PLEASE INPUT
          THE PLANE No. ABOUT WHICH THE UNBALANCED PRIMARY COUPLE
          DUE TO RECIPROCATING MASSES IS TO BE DETERMINED
          11
          THE PLANE No. ABOUT WHICH THE UNBALANCED SECONDARY COUPLE
          DUE TO RECIPROCATING MASSES IS TO BE DETERMINED
          11
```

7. OUTPUT

The program gives first the pressure–θ curve. This is not shown here to save space. Figure 4 shows the output of the residual inertia force of an individual cylinder of the engine. The program then gives the crankpin, wristpin and main bearing loads, the first two of which are given here in Figs 5 and 6, respectively. The engine torque–θ curve with the fluctuation of energy is next shown by the program, as in Fig. 7. These figures can be printed out by using the Print Screen command. (Figs 5–7 are actual prints, which are distorted with circles appearing as ellipses, because of graphics problems.)

The output file TABLE.PAS, consists of (1) bearing loads, (2) engine torque, (3) cylinder torque, all in tabular form at 5° intervals, (4) flywheel inertia, (5) primary and secondary unbalance forces of the engine, (6) rotating parts balancing masses in flywheel and damper planes and (7) primary and secondary unbalance moments of the engine. The output in items (1)–(3) is not given here to save space. The rest of the output is given below.

```
THE MEAN TORQUE OUTPUT FOR 9 CYLINDER IS 11939.47 Nm

INERTIA OF 9 CYLINDER, DAMPER AND GENERATOR = 264.35

MOMENT OF INERTIA TO KEEP THE SPEED FLUCTUATION WITHIN DESIRED
    LIMIT = 41.07
```

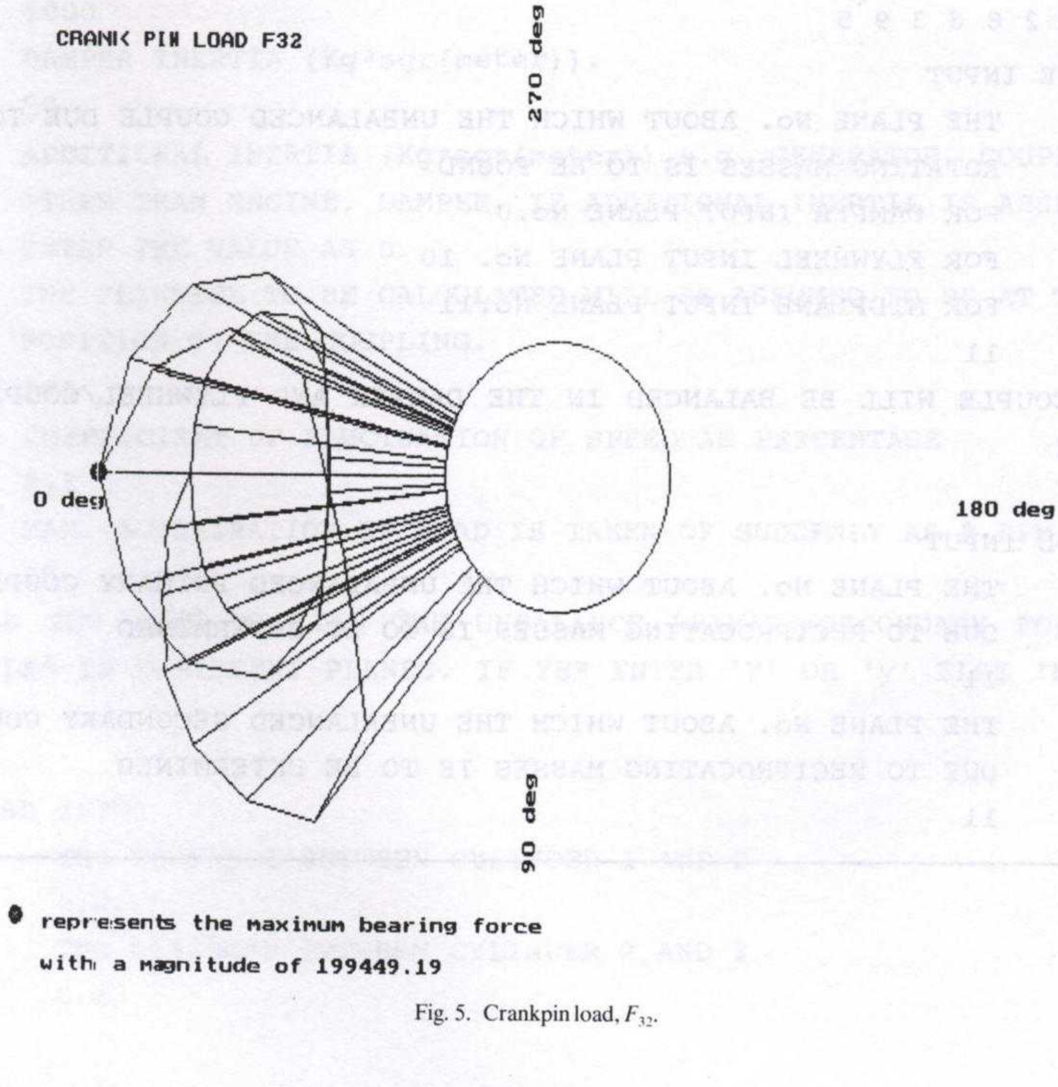

Fig. 5. Crankpin load, F_{32}.

```
MOMENT OF INERTIA TO KEEP THE MAX. PERMISSIBLE ACC. WITHIN
    THE DESIRED LIMIT = 456.05

THE SYSTEM AS SUCH HAS MORE INERTIA THAN IS REQUIRED TO KEEP
    THE COEFFICIENT OF FLUCTUATION OF SPEED WITHIN 0.10 PERCENT

THE DESIRED MOMENT OF INERTIA OF FLYWHEEL ON THE BASIS OF MAX.
    ACCELERATION ALLOWED = 191.70 Kg*sqr(meter)

THE UNBALANCED PRIMARY FORCE IS 2.44 Newtons
THE ANGULAR LOCATION OF UNBALANCED PRIMARY FORCE IS 320.00

THE UNBALANCED SECONDARY FORCE IS 0.80 Newtons
THE ANGULAR LOCATION OF UNBALANCED SECONDARY FORCE IS 280.00

THE UNBALANCED COUPLE DUE TO ROTATING PARTS ABOUT MIDPLANE IS
    12127.51 Nm
```

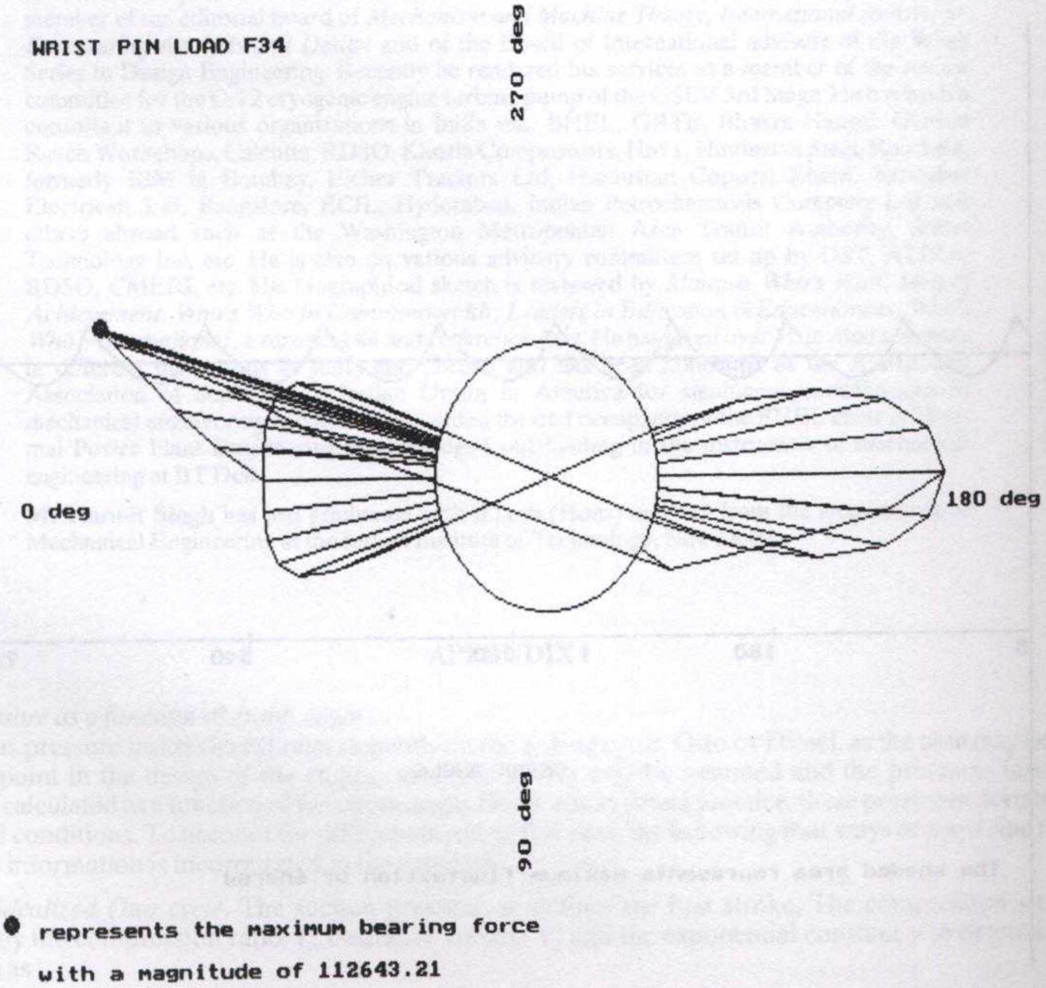

Fig. 6. Wristpin load, F_{34}.

THE BALANCING Kg.mm. VALUE IN FLYWHEEL/COUPLING PLANE IS GIVEN
BY 257.16

THE ANGULAR POSITION WRT CYLINDER 1 CRANK IS 349.998

THE BALANCING Kg.mm. VALUE IN DAMPER PLANE IS GIVEN BY 257.21

THE ANGULAR POSITION WRT CYLINDER 1 CRANK IS 169.990

THE UNBALANCED PRIMARY COUPLE IS 39414.42 Nm

THE UNBALANCED SECONDARY COUPLE IS 12087.69 Nm

8. CONCLUSION

The above output has been compared with manual calculations for the same problem and there is complete agreement between them. To validate the program further, it has also been checked with the worked-out problem given in [1] (Tables 13.1–13.4, Figs 13.15, 13.16 and 14.3). Thus, this program can be used by students to understand the complete reciprocating machine installation dynamics problem. This program can also be used by engineers in industry for determining the dynamics quantities required in the design of reciprocating machine installations.

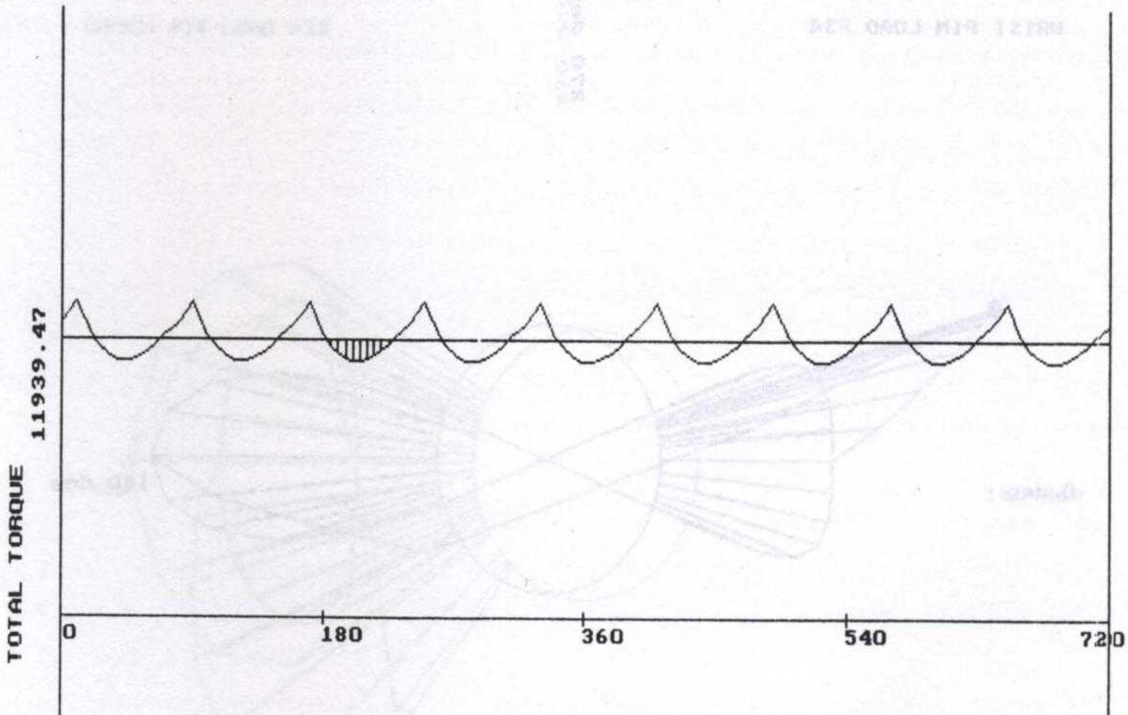

Fig. 7. Engine torque–θ curve, with the fluctuation of energy.

REFERENCES

1. J. S. Rao and R. V. Dukkipati, *Mechanism and Machine Theory*, Wiley Publications (1989). (A revised and enlarged edition with a solution manual is due to appear.)

Professor J. S. Rao graduated from Andhra University with first class honours in 1960, did his Masters degree in 1963, PhD in 1965 and DSc in 1971 from IIT Kharagpur. After doing post-doctoral research and teaching at the University of Surrey, Dr Rao was appointed as professor in Kharagpur and moved to IIT Delhi in 1975 to start a new centre for Industrial Tribology, Machine Dynamics and Maintenance Engineering. Subsequently, he taught at the National Institute of applied Sciences, Lyon, France; Concordia University, Montreal, Canada; Rochester Institute of Technology and Gesamthochschule in Kassel, Germany, as Visiting Professor. He is currently at National Chung Cheng University, Chia-Yi, 621 Taiwan. Professor Rao served as Science Counsellor at the Indian Embassy in Washington on deputation and has been responsible for various projects and schemes initiated during 1984–1988. Professor Rao is a member of the Indian National Academy of Engineering, American Society of Mechanical Engineers, Indian Society of Theoretical and Applied Mechanics, Association of Machines and Mechanisms, Indian Society of Technical Education, etc., and was Founder President of the Association of Machine and Mechanisms from 1984 to 1988 and President of the Indian Society of Theoretical and Applied Mechanics from 1990 to 1992. He has been the chairman of a 12-nation committee on rotor dynamics set up by the International Federation of Theory of Machines and Mechanisms (IFToMM) from 1983 to 1990 and responsible for the organization of international conferences and technical meetings to discuss the state of art in the field of rotor dynamics. He has been a member of the IFToMM executive council since 1984 and Chairman of the IFToMM Permanent Commission on Conferences since 1990. He is currently chairman of the Rotor Dynamics Group set up by the Aeronautical Research and Development Board and a member of the Structures and Dynamics Committee set up by the International Gas Turbine Institute of the American Society of Mechanical Engineers. Professor Rao has produced 26 PhDs and published five textbooks, *Rotor Dynamics*, *Theory and Practice of Mechanical Vibration*, *Turbomachine Blade Vibration*, *Mechanism and Machine Theory* and *Advanced Theory of Vibration* with John Wiley and 220 papers. He is a member of the editorial board of *Mechanism and Machine Theory*, *International Journal for Engineering Analysis and Design* and of the Board of International advisers of the Wiley Series in Design Engineering. Recently he rendered his services as a member of the review committee for the C-12 cryogenic engine turbine pump of the GSLV 3rd Stage. He has been a consultant to various organizations in India viz., BHEL, GRTE, Bhakra Nangal, Garden Reach Workshops, Calcutta, RDSO, Khosla Compressors, HMT, Hindustan Steel, Rourkela, formerly IBM in Bombay, Eicher Tractors Ltd, Hindustan Copper, Khetri, Kirloskar Electricals Ltd, Bangalore, ECIL, Hyderabad, Indian Petrochemicals Company Ltd and others abroad such as the Washington Metropolitan Area Transit Authority, Stress Technology Inc, etc. He is also on various advisory committees set up by DST, ATIRA, RDSO, CMERI, etc. His biographical sketch is reviewed by *Marquis Who's Who*, *Men of Achievement*. *Who's Who in Commonwealth*, *Leaders in Education &Educationists*, *Who's Who—International*, *Learned Asia* and *Reference Asia*. He has given over 70 invited seminars in different institutions in India and abroad and has been honoured as the ASME and Association of Scientists of Indian Origin in America for significant contributions in mechanical engineering. He was also awarded the first occupancy of the BHEL chair in Thermal Power Plant Engineering and adjudged outstanding in the instruction of mechanical engineering at IIT Delhi.

Mr Harmit Singh has just graduated with BTech (Hons) in 1993 from the Department of Mechanical Engineering at the Indian Institute of Technology, New Delhi.

APPENDIX I

Gas pressure as a function of crank angle

The gas pressure inside the cylinder depends on the engine cycle, Otto or Diesel, as the case may be. As a starting point in the design of the engine, idealized cycles may be assumed and the pressures inside the cylinder calculated as a function of the crank angle. However, in actual practice, these pressures deviate from idealized conditions. To account for different forms of this data, the following four ways of providing the gas pressure information is incorporated in the program.

A.1. Idealized Otto cycle. The suction pressure, p_s defines the first stroke. The compression stroke is defined by the compression ratio, r_v, clearance volume V_c and the exponential constant γ to determine the pressure as

$$p(\theta)\,V(\theta)^{\gamma} = p_s(V_c + V_{\text{stroke}})^{\gamma}.$$

At the end of compression, the pressure rises instantaneously from $p(\theta = 360°)$, determined from the above, to a value specified in the input, as pressure after combustion p_c (the output shows the value of p at $\theta = 360°$ as obtained from the equation above and the value at $\theta = 365°$ takes into account the sudden pressure rise at $\theta = 360°$). The user can also provide the pressure ratio for the rise in this period taken in the program as 5° for convenience.

The expansion stroke is given by the following relation.

$$p(\theta)\, V(\theta)^{\gamma} = p_c V_c^{\gamma}.$$

At the end of expansion stroke at $\theta = 540°$, the pressure inside the cylinder drops suddenly to the suction pressure and that pressure remains same during the exhaust stroke at p_s.

A.2. Idealized Diesel cycle. In the Diesel cycle, the suction and compression strokes remain same from $\theta = 0°$ to 360° as in the Otto cycle. The pressure during the combustion remains same from $p(360° \leqslant \theta \leqslant \theta_e)$, where θ_e is determined from the percentage stroke at which expansion begins (given in the input. The expansion and exhaust remain same for the remainder of the cycle.

A.3. Practical cycle. In this case, the suction and exhaust pressures are different and given as inputs. The opening and closing of the inlet valve respectively from the head end dead centre and crank and dead centre and, similarly, the opening and closing of the exhaust valve respectively from the crank end dead centre and head end dead centre are also specified in the input. The percentage of the stroke at which the maximum pressure is achieved and at which expansion begins, along with the compression ratio are also to be specified. The last quantity in the input is the compression and expansion exponent.

A.4. Pressure as a function of crank angle. Quite often, the pressures in an existing engine are recorded in an indicator diagram. For the purpose of analyzing such a 2- or 4-stroke engine installation, the program provides for the pressures to be input at 5°, or a multiple of 5°, intervals of the crank. If the input is given in multiples of 5°, the pressure is determined by linear interpolation for all 5° positions of the crank and given in the output.