Computer-aided analysis of centrifugal compressors

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This paper describes computer-aided analysis of centrifugal compressors (CAACC), a micro-computer-based, interactive, and menu-driven software package for use as an educational tool by mechanical engineering students studying radial flow compressors. CAACC is written in the Pascal computer language and runs on IBM PC, or compatible, computers. In addition to solving for any unknown variables, the graphical utilities of the package allow the user to display a diagrammatic sketch of the compressor and to draw velocity diagrams at several locations. Furthermore, the program allows the investigation and plotting of the variation of any parameter versus any other parameter. Through this option, the package guides the student in learning the basics of centrifugal compressors by the various performance studies that can be undertaken and graphically displayed. The comprehensive example presented demonstrates the capabilities of the package as a teaching tool.

NOTATION

C_p	specific heat at constant pressure
C_w	whirl component of velocity
P_{01}	inlet stagnation pressure
P_{03}	exit stagnation pressure
T_{01}	inlet stagnation temperature
T_{03}	exit stagnation temperature
U	impeller tip speed
U_e	average speed of impeller eye
U_{er}	speed of impeller eye root
U_{et}	speed of impeller eye tip
W_c	compressor work
σ	slip factor
ψ	power input factor

INTRODUCTION

The centrifugal compressor is usually introduced to mechanical engineering students in a senior course which concentrates on the application of thermodynamics to gas turbines. No attempt will be made here to fully describe it; rather, a brief description will be given in the next section. For further details of the subject the reader is referred to textbooks [1–7].

A large number of parameters affect the performance of centrifugal compressors and, depending on the known variables, the inter-parametric relations may not be given via simple functions. In such cases, one has to solve nonlinear equations in order to obtain the values of the parameters in question. Performing this task by hand is time-consuming. This limits the number of cases that can be considered by students and prevents them from exploring the effects of design parameters on compressor performance.

With advances in computer technology, microcomputers are not only becoming more and more reliable in both speed and accuracy, but are also now within the reach of the majority of engineering students. Therefore, if given the tool, students may exploit microcomputers in solving time-consuming problems. Through their use, students can solve more problems and conduct more analyses of a given type than when using hand-calculation. This in turn, leads students into a deeper understanding of the basic principles governing a particular application.

The aim of this paper is to present a program developed to provide the mechanical engineering student with a tool that allows him to explore the effects of design changes on the performance of centrifugal compressors without the boredom of performing hand-calculations.

PRINCIPLE OF OPERATION AND THEORETICAL PERFORMANCE

The centrifugal compressor consists of an impeller rotating within a stationary casing. During operation, gas is sucked into the impeller eye and whirled round at high speed by the vanes of the impeller disk. While flowing, the fluid receives energy from the vanes resulting in an increase in both pressure and absolute velocity. At any point in the flow through the impeller, the centrifugal acceleration is obtained by a pressure head, so that the static pressure of the gas increases from the eye to the tip of the impeller. The remainder of the static pressure rise is obtained in the volute casing surrounding the impeller or in flow through fixed diverging passages known as the diffuser, in which, the very high velocity of the fluid leaving the impeller tip is reduced to somewhere in the region of the velocity with which it enters the impeller eye. Friction in the diffuser causes some loss in stagnation pressure. Unless otherwise specified, CAACC performs computations assuming half of the pressure rise occurs in the impeller and the other half in the diffuser.

Since no work is done on the gas in the diffuser, the energy absorbed by the compressor is determined by the conditions of the fluid at inlet and outlet of the impeller. Designating the conditions at impeller inlet, impeller exit, and compressor outlet by subscripts 1, 2, and 3 respectively, the work absorbed by the compressor and the stagnation pressure rise of the gas across the compressor are given by

$$W_c = \psi(\sigma U^2 - C_{w1}U_e) = C_p(T_{03} - T_{01}) \tag{1}$$

$$\frac{P_{03}}{P_{01}} = \left[1 + \frac{(T_{03} - T_{01})}{T_{01}}\right]^{\gamma/(\gamma-1)} \tag{2}$$

where

$$U_e = \frac{U_{er} + U_{et}}{2} \tag{3}$$

and ψ, C_p, U, C_{w1} and σ being the power input factor, the specific heat, the blade tip velocity, the whirl velocity component at inlet, and the slip factor respectively.

Usually, the stagnation conditions (ambient conditions when the working fluid is air) are known at the compressor inlet. The package exploits this information and starts iterations, whenever needed, using these values as an initial guess in order to compute the inlet axial velocity and static conditions. Further, in calculating the required depth of the impeller channel at the periphery, some assumptions regarding the radial component of velocity at the tip and the division of losses between the impeller and the diffuser must be made in order for the density to be evaluated. The usual design practice is to choose the radial component of velocity equal to the axial velocity at inlet to the eye. Unless otherwise instructed, the program follows this practice. In addition, if the losses in the impeller and diffuser are not specified, the package divides the total loss equally between them. Once the radial component of velocity at impeller tip and the isentropic efficiency of the impeller are known, all fluid properties at that location are found by using the isentropic relations.

The current trend in compressor technology is towards high-performance compressors using backswept curved vanes. The use of such vanes in the impeller imply less stringent diffusion requirements in both the impeller and diffuser, tending to increase the efficiency of both components and to widen the compressor operating range. CAACC permits the user to explore the effects of changing the vane backswept angle on the overall performance of the compressor.

After leaving the impeller and before entering the diffuser, gas flows in a vaneless space. In this gap, no energy is supplied to the fluid whose angular momentum remains constant (neglecting friction). From this condition, the whirl component of the velocity at diffuser inlet is determined. The radial component of the velocity is calculated from the continuity equation using a guess-and-correct procedure. The velocity decreases during flow in the vaneless space and as such some diffusion takes place. This is to advantage since it decreases the amount of diffusion needed in the diffuser. Knowing the velocity components at diffuser inlet, the angle of the diffuser vane leading edge is easily determined. The throat width of the diffuser is calculated by an iterative procedure using continuity and isentropic relations. For any required outlet velocity, knowing the total loss for the whole compressor, the final density is easily calculated and hence the flow area required at outlet. The length of the passage after the throat then depends on the chosen angle of divergence.

DESCRIPTION OF THE PROGRAM

CAACC is written in Pascal using the Turbo Pascal compiler version 6 [8] which includes a powerful debugger and facilitates enhanced graphics options. The package runs on any IBM PC XT/AT or compatible which includes at least 640 Kbytes of main memory, a colour monitor equipped with a *V*ideo or *E*nhanced *G*raphics *A*daptor card (VGA or EGA), and an optional maths coprocessor.

The software is divided into three major modules. The first one consists of procedures for generating menus and data entry windows. The role of the second module which is composed of several procedures, is to solve for unknowns. An iterative solution procedure is adopted. The iterative process consists of repeatedly executing the solving routine until no further changes in the number of known variables is noticed. The purpose of the third module is to view, print, and plot results. The second and third modules are intercorrelated owing to the need for repeatedly solving for unknowns while plotting the variation of one quantity versus another. The program allows the plotting of the variation of three quantities at a time.

The menu structure of the program, depicted in Figs 1 and 2, is designed in such a way as to provide a user-friendly environment. The main menu offers five entries. By selecting the appropriate task from this menu, one can input data, solve for unknowns, optimize the compressor performance, or use the graphics screen as a window through which a schematic of the compressor or of velocity diagrams may be seen or printed. The user starts by choosing entry INPUT which allows

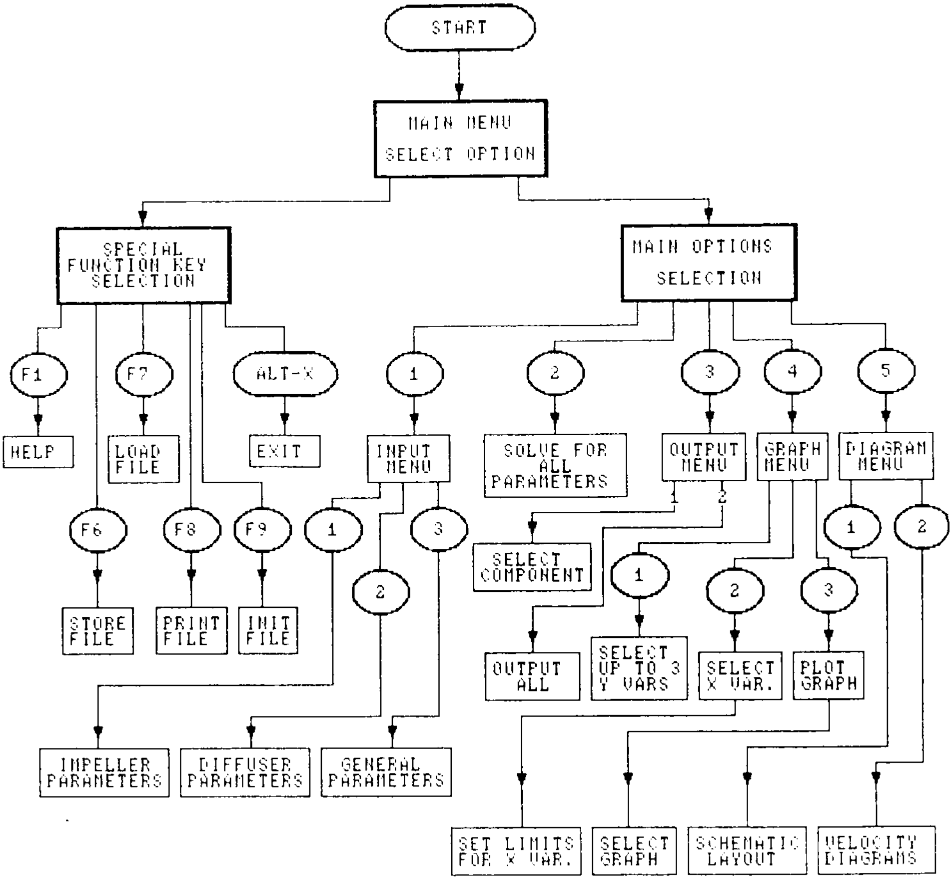

Fig. 1. Hierarchical menu structure of CAACC.

them to enter the values of the known quantities. As depicted in Fig. 2, the menu-driven structure of the package facilitates this task by guiding the user throughout the data-entering procedure. Having entered all data, the user may store them into a file that can be read by the program at any time. This option is included to permit easy correction of wrongly inputted or missing data. After inputting all data, a solution is obtained by choosing the SOLVE entry. Results may be viewed and printed by going to the OUTPUT entry (Fig. 2). The effect of varying a parameter on the overall performance of the compressor is studied as shown in Figs 1 and 2, in the GRAPH entry. This allows a comprehensive analysis of the different parameters involved. A diagrammatic sketch of the compressor and velocity diagrams at several locations may be seen from the DIAGRAM entry. To increase the usefulness of the package and to facilitate its usage, a HELP option, accessable from any level during execution, is included. Hard copies of the compressor and velocity diagrams may be obtained by using the 'print-screen ' command with a suitable graphics printer via any memory-resident IBM graphics module.

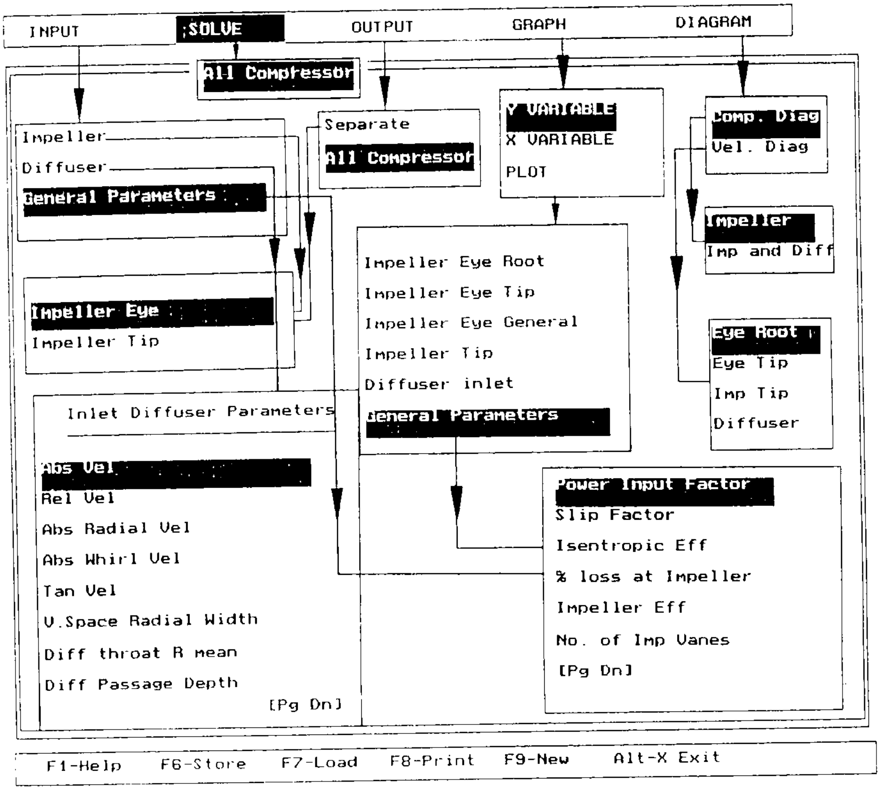

Fig. 2. Main menu and several illustrative submenus of CAACC.

REPRESENTATIVE GRAPHICAL PLOTS

The capabilities of the program are demonstrated in this section by presenting a comprehensive study of a test problem. The centrifugal compressor in question is considered to be part of an aircraft engine. It is required to determine the performance of the compressor at take-off (sea-level altitude and zero forward velocity). Records of input and output data are presented in Table 1, while graphical output results are shown in Fig. 3. Figs 4 to 7 reveal the compressor responses to variation in its parameters.

In Fig. 3, the velocity triangles at impeller eye root (Fig. 3(a)), impeller eye tip (Fig. 3(b)), impeller tip (Fig. 3(c)), and diffuser inlet (Fig. 3(d)) are presented. The variation of the various velocities as gas flows through the compressor are easily depicted.

The effect of varying any parameter on the overall performance of the compressor may be studied through the GRAPH option. Such analysis was undertaken here and results are presented in Figs 4 to 7.

The influence of compressor isentropic efficiency on the overall pressure ratio and diffuser inlet Mach number is depicted in Figs 4(a) and 4(b) respectively. As expected, the overall pressure ratio increases with increasing the isentropic efficiency of the compressor owing to the decrease in stagnation pressure loss. The decrease in the diffuser inlet Mach number is attributed to an increase in the static temperature and a decrease in the absolute velocity. In Figs 5(a), 5(b), 6(a), and 6(b)

Input data

```
====================================

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====================================


EYE ROOT PARAMETERS
===================

Eye root Radius          :0.075

EYE TIP PARAMETERS
==================

Eye tip Radius           :0.150

EYE GENERAL PARAMETERS
======================

stag Temp                :295.000
stag Pressure            :1.100
Prewhirl Angle           :0.000

IMPELLER TIP PARAMETERS
=======================

Imp Tip Radius           :0.250

DIFFUSER INLET PARAMETERS
=========================

V.Space Radial Width     :0.050
Diff. Throat R mean      :0.330
Diff passage Depth       :0.018
No. of Diff Vanes        :12.000
```

```
EYE GENERAL PARAMETERS
======================

stat Temp                :284.896
stag Temp                :295.000
stat Pressure            :0.974
stag Pressure            :1.100
Mean Radius              :0.112
Prewhirl Angle           :0.000

IMPELLER TIP PARAMETERS
=======================

Abs Vel                  :434.057
Rel Vel                  :149.662
Abs Radial Vel           :142.561
Abs Whirl Vel            :409.978
Tan Vel                  :455.531
Imp Tip Radius           :0.250
Imp Width                :0.017
Angle(Vabs,Tan)          :19.174
Backswept Angle          :17.720
stat Temp                :394.528
stag Temp                :488.262
stat Pressure            :2.604
stag Pressure            :5.491

DIFFUSER INLET PARAMETERS
=========================

Abs Vel                  :355.024
Rel Vel                  :226.580
Abs Radial Vel           :96.531
Abs Whirl Vel            :341.648
Tan Vel                  :546.637
V.Space Radial Width     :0.050
Diff. Throat R mean      :0.330
Diff passage Depth       :0.018
```

Table 1. Input and output data for a centrifugal processor.

```
GENERAL PARAMETERS
==================

Power Input Factor          :1.040
Slip Factor                 :0.900
Isentropic eff              :0.780
R.V.S - N                   :290.000
Air mass flowrate           :9.000
```

Output data

```
===================================

            G.T.C.A.
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===================================


EYE ROOT PARAMETERS
===================

In Abs Vel              :142.561
In Rel Vel              :197.483
In Abs Axial Vel        :142.561
In Abs Whirl Vel        :0.000
Tan Vel                 :136.659
Eye root Radius         :0.075
Angle(Vane,Tan.)        :46.211

EYE TIP PARAMETERS
==================

In Abs Vel              :142.561
In Rel Vel              :308.264
In Abs Axial Vel        :142.561
In Abs Whirl Vel        :0.000
Tan Vel                 :273.319
Eye tip Radius          :0.150
Angle(Vane,Tan.)        :27.546
```

```
No. of Diff Vanes          :12.000
Diff vanes Inlet Angle:15.777
Throat Width               :0.043
stat Temp                  :425.554
stag Temp                  :488.262
stat Pressure              :3.394
stag Pressure              :5.491

GENERAL PARAMETERS
==================

Power Input Factor              :1.040
Slip Factor                     :0.900
Isentropic eff                  :0.780
% loss at impeller              :50.000
Impeller eff                    :0.890
No of Imp Vanes                 :20.000
Imp tip Pressure Ratio          :4.992
Overall Pressure Ratio          :4.241
Overall change in Tstag.        :193.262
R.V.S - N                       :290.000
Air mass flowrate               :9.000
Theo Torque                     :102.494
Theo Work                       :186.758
Actual Work                     :194.228
Rel Imp in Mach #               :0.911
Imp tip Mach #                  :1.090
Rad Imp tip Mach #              :0.358
Diff inlet Mach #               :0.859
```

Table 1. Input and output data for a centrifugal processor.

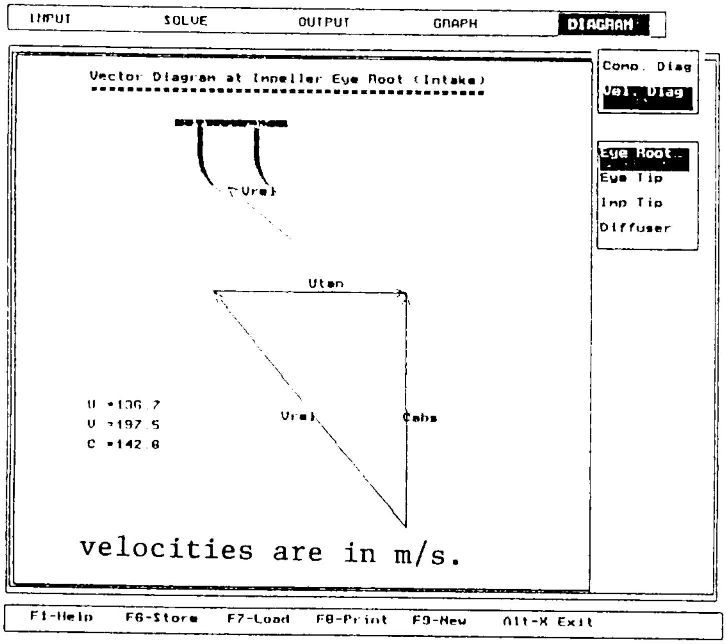

Fig. 3(a). Velocity diagram at impeller eye root.

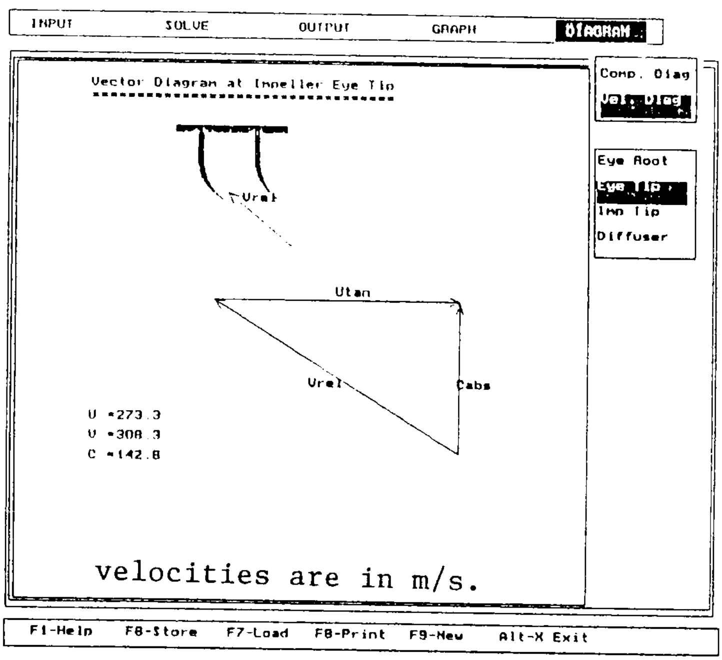

Fig. 3(b). Velocity diagram at impeller eye tip.

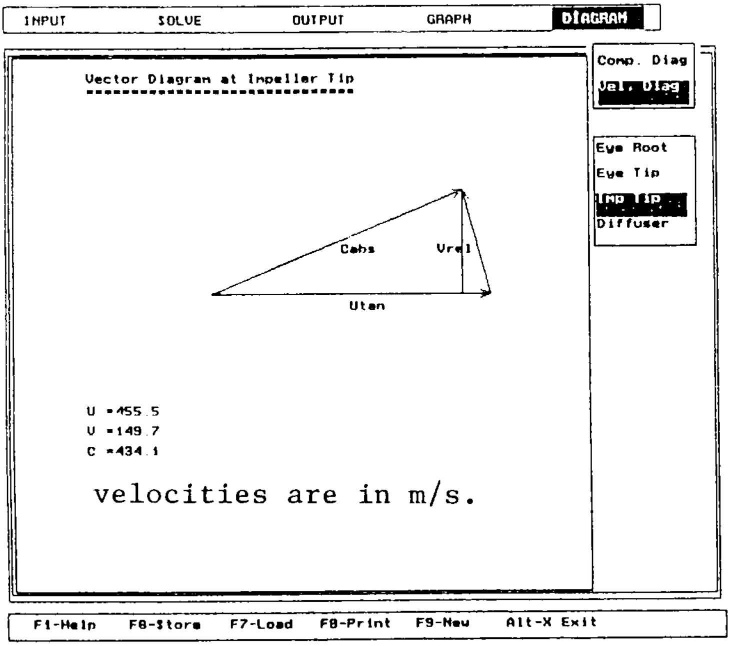

Fig. 3(c). Velocity diagram at impeller tip.

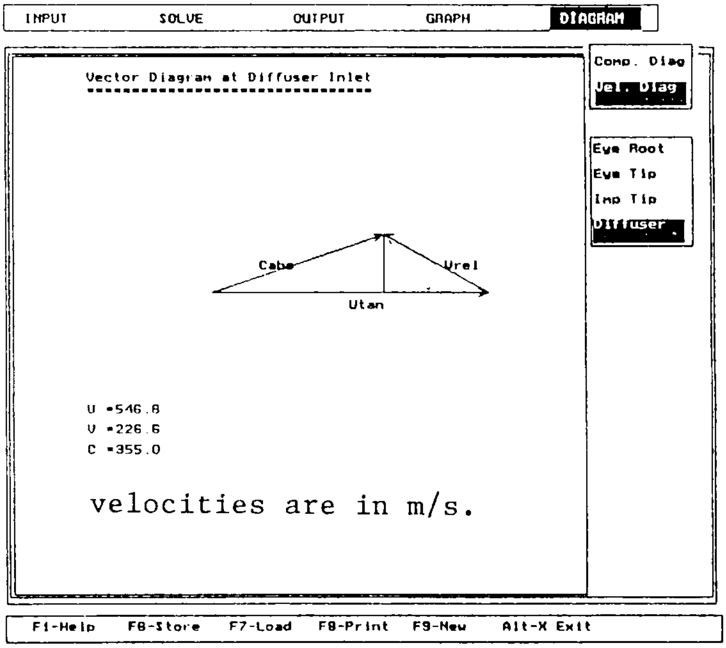

Fig. 3(d). Velocity diagram at diffuser inlet.

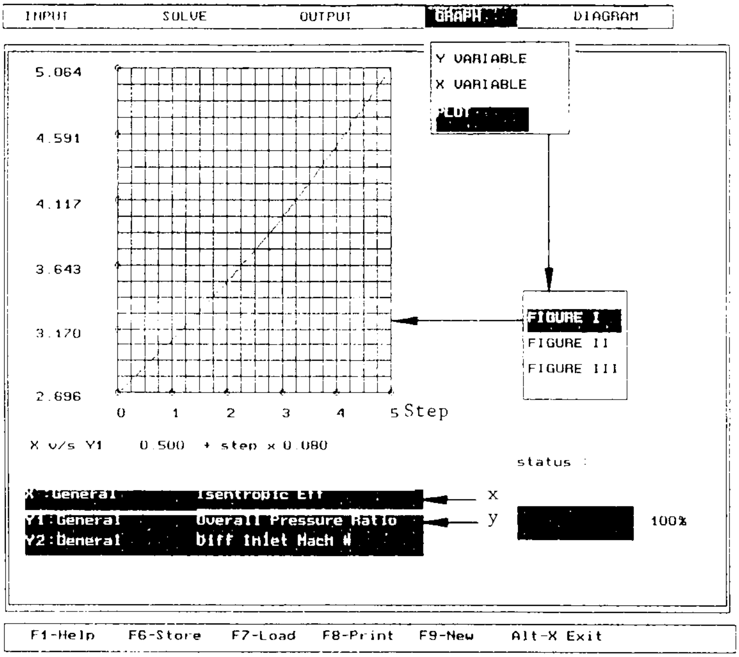

Fig. 4(a). Variation of overall pressure ratio with isentropic efficiency.

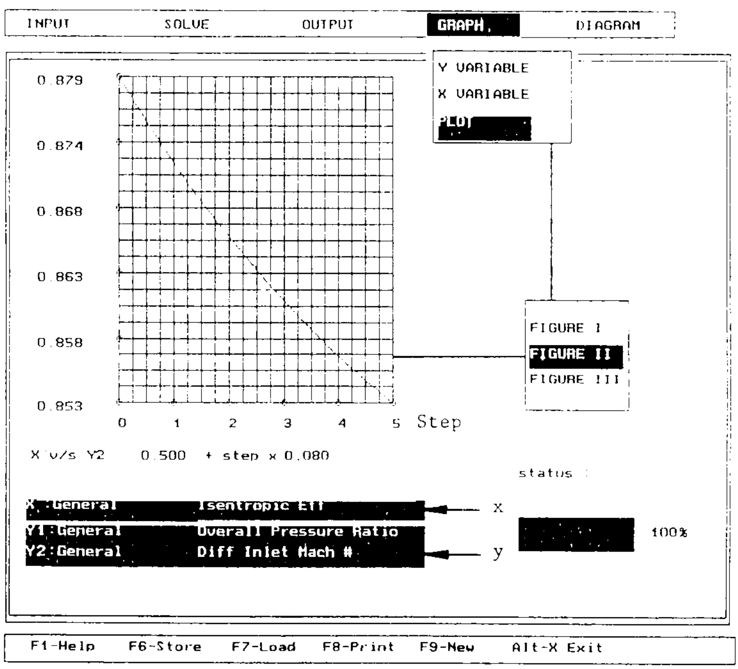

Fig. 4(b). Variation of diffuser inlet Mach number with isentropic efficiency.

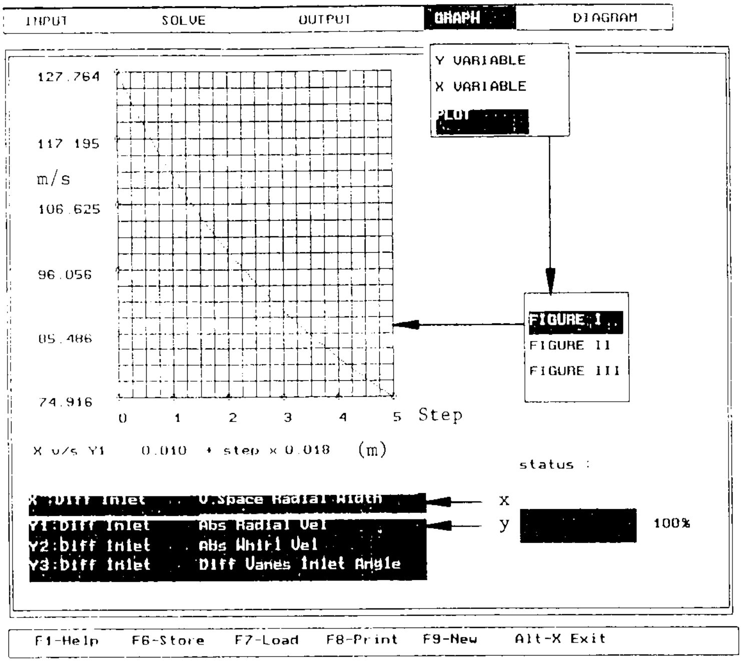

Fig. 5(a). Variation of radial velocity at diffuser inlet with vaneless space radial width.

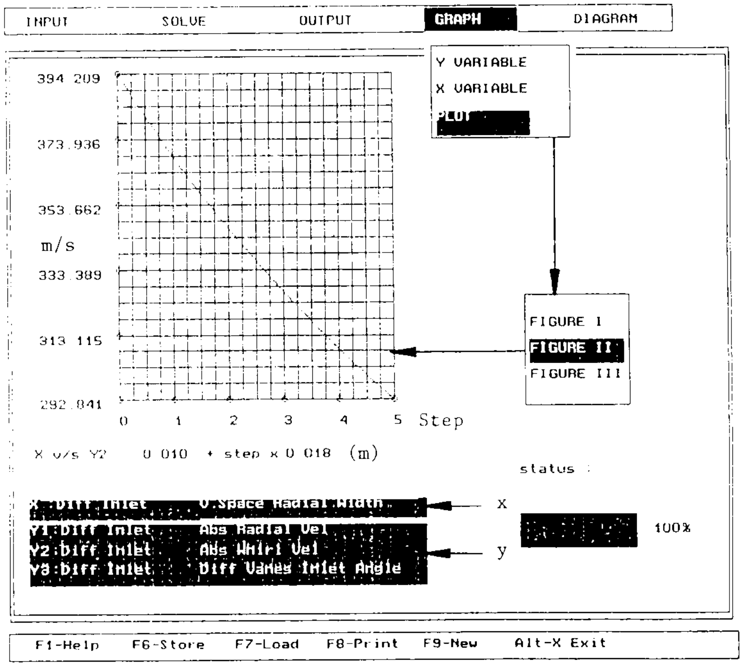

Fig. 5(b). Variation of the whirl velocity at diffuser inlet with vaneless space radial width.

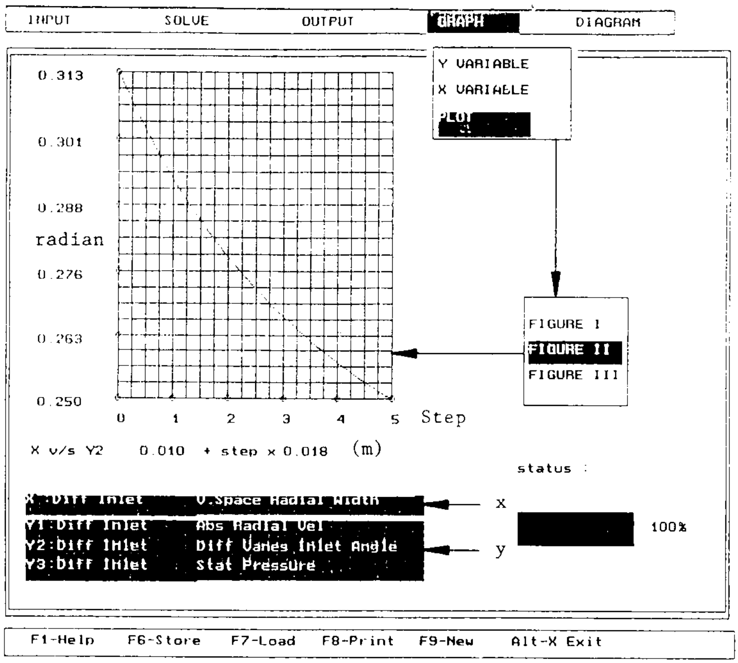

Fig. 6(a). Variation of the diffuser vanes inlet angle with vaneless space radial width.

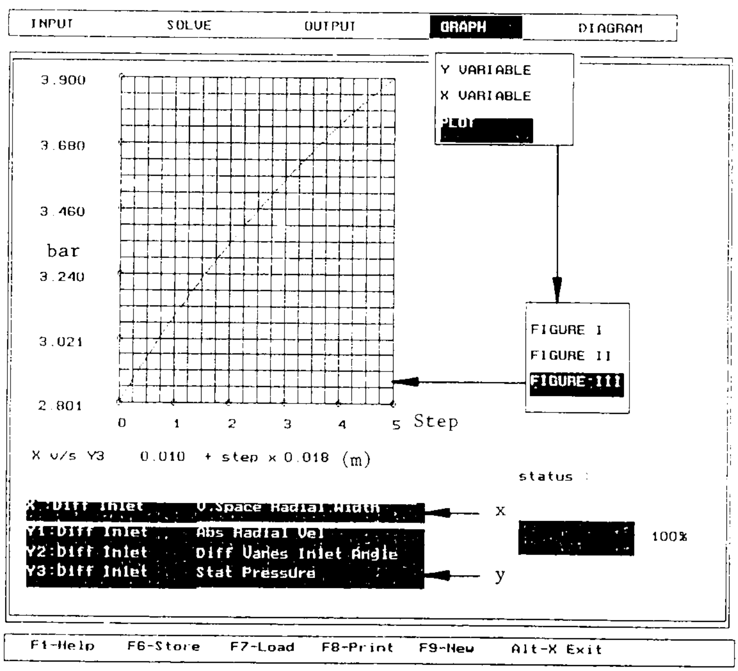

Fig. 6(b). Variation of the air static pressure at diffuser inlet with vaneless space radial width.

the variations of the radial velocity component, the whirl velocity component, the air angle, and the air static pressure at diffuser inlet with the vaneless space radial width are respectively shown. In accordance with continuity, velocity and air angle are decreased with increasing values of the vaneless space. Furthermore, since more diffusion is taking place, owing to the decrease in velocity, the static pressure should increase. This is manifested by the results presented in Fig. 6(b).

Finally, the effects of varying the pre-whirl angle on the compressor actual work and the impeller tip Mach number are presented in Figs 7(a) and 7(b) respectively. As can be seen, the work capacity of the compressor decreases with increasing the pre-whirl angle (Fig. 7(a)). Furthermore, increasing the pre-whirl angle decreases both the impeller tip static temperature and absolute velocity. The rate of decrease in the static temperature, however, is higher than the rate of decrease in velocity. The net effect is an increase in the impeller tip Mach number (Fig. 7(b)).

CONCLUSION

A microcomputer-based program for analysing centrifugal compressors was developed. The menu-driven structure of the program, along with its HELP utility and enhanced graphical capabilities, allow the mechanical engineering student to easily explore the effect of changing design parameters on compressor performance. The package also provides the user with an efficient educational tool on a low-cost workstation. The capabilities of the package as a teaching tool are demonstrated by presenting a comprehensive analysis of an illustrative example problem. If hand-calculations are combined with this educational tool, the user will be able to solve problems more complicated than the one presented in this paper very easily. Copies of the package will be provided to users on request to the authors.

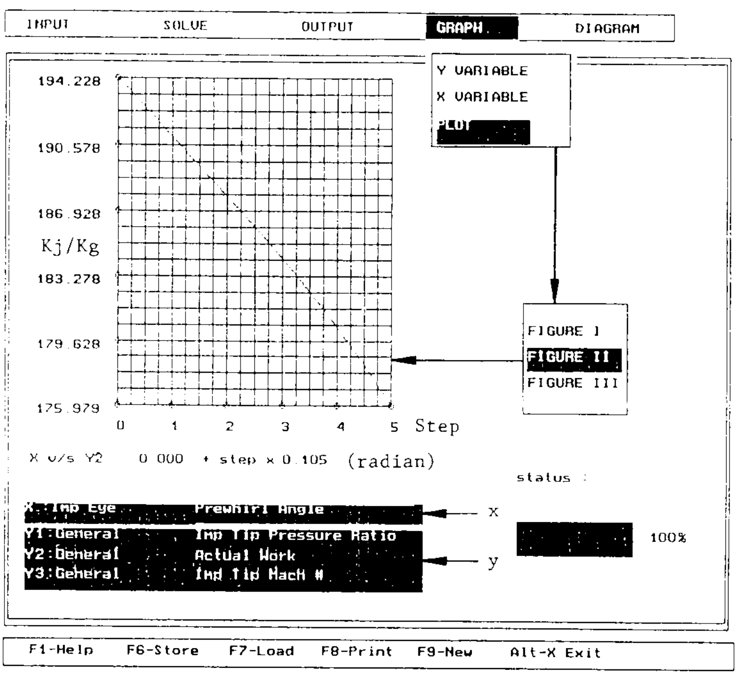

Fig. 7(a). Variation of the compressor actual work with pre-whirl angle.

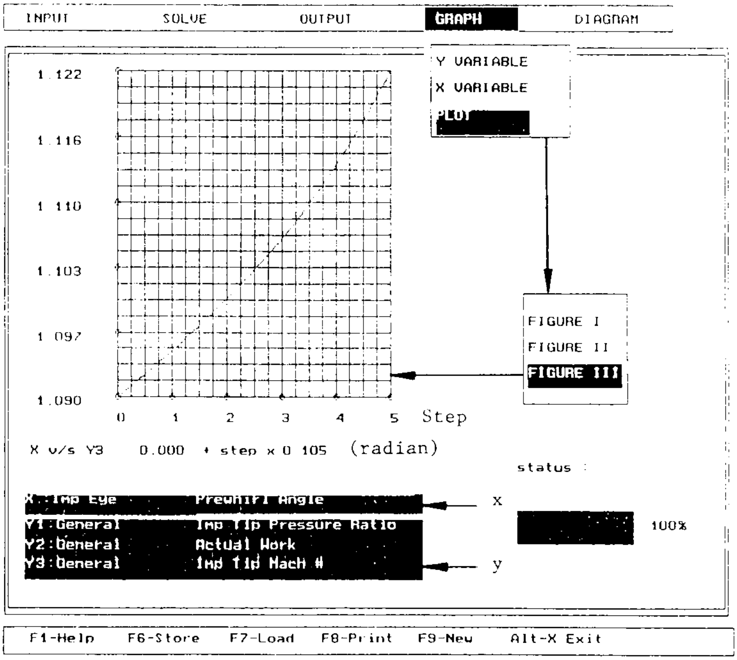

Fig. 7(b). Variation of impeller tip Mach number with pre-whirl angle.

ACKNOWLEDGEMENTS

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