

Computer-aided analysis of hydraulic reaction turbines

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A microcomputer-based educational software package designed to help mechanical engineering students to understand hydraulic reaction turbines is described. The software is interactive, menu-driven, and easy-to-use, is written in the Pascal computer language, and runs on IBM PC, or compatible, computers. The program can handle radial, mixed, or axial flow turbine problems by solving for unknown variables through a complete set of equations covering the turbine installation. Using similarity laws, model-prototype problems or operation under different conditions can also be tackled. Furthermore, the program is equipped with graphical utilities that include many diagrammatic sketches of reaction turbines, some recommended charts, and the possibility of drawing velocity triangles when corresponding variables are available. The most important feature of the package is an option that allows one to plot the variation of any parameter versus any other one. Through this option, the student can easily understand and discuss the effects of varying design parameters on the overall performance of the machine. Finally, some special features that are important in making the package user-friendly and encouraging-to-use are also available, and the comprehensive example problem provided demonstrates the capabilities of the package as an instructional tool.

NOMENCLATURE

bp	brake power (W)
h	net head (m)
h'	effective head available at runner inlet (m)
h''	extracted head from water that is transmitted to the runner (m)
H_1	total energy head at runner inlet (m)
H_2	total energy head at runner exit (m)
H_B	total energy head at inlet to the scroll case (m)
H_C	total energy head at tailrace (m)
η_h	hydraulic efficiency
n_s	specific speed ($\text{N m}^{-3/2} \text{ s}^{-3})^{1/2}$)
P_{atm}	atmospheric pressure (N/m^2)
pe	water power at runner exit (W)
pf_2	hydraulic power loss (W)
pis	power input to shaft (W)

P_v	water vapour pressure (N/m^2)
rpm	rotative speed
rpm_e	rotative speed at maximum efficiency
Q	volume flow rate flowing in runner (m^3/s)
r_1	radius of runner at which water enters (m)
r_2	radius of runner at which water leaves (m)
T	torque input to shaft (N m)
u_2	runner velocity at exit (m/s)
V_1	absolute water velocity at runner inlet (m/s)
V_2	absolute water velocity at runner exit (m/s)
v_2	relative water velocity at runner exit (m/s)
Va_2	axial water velocity (m/s)
Vr_g	radial water velocity at exit from the guide vanes (m/s)
Vt_g	tangential water velocity at exit from the guide vanes (m/s)
Vu_1	tangential water velocity at runner inlet (m/s)
Vu_2	tangential water velocity at runner exit (m/s)
z_B	elevation of the scroll case assembly above tailwater (m)
z_{Bmax}	maximum elevation of the scroll case assembly above tailwater (m)
α_1	angle between V_1 and u_1 (degree)
α_2	angle between V_2 and u_2 (degree)
α_g	direction angle of the guide vanes (degree)
β_1	inlet angle of runner vane (degree)
β_2	exit angle of runner vane (degree)
γ	specific weight (N/m^3)
ρ	density (kg/m^3)
σ	cavitation parameter
σ_C	critical cavitation parameter

INTRODUCTION

The recent rapid evolution in the development of reliable and cheap microcomputers has broadened their accessibility and utilization and has influenced the teaching/learning approach in almost all disciplines. Research has shown that [1] students from all disciplines studying at their own pace using student-oriented computer programs demonstrate increased motivation which translates into improved achievement of course objectives. In the area of engineering curricula, the use of computer-based learning tools not only allows students to learn difficult concepts by interactive dialogues, but also offers the possibility of training them in problem-solving skills. This is achieved by designing easy-to-use programs [2–7] that allow students to tackle a wide range of applications of a certain principle by employing various types of problem-solving approaches. Therefore, teaching methods using personal computers seem to best complement an engineering education.

Hydraulic turbomachinery is an important area within mechanical engineering that deals with the design and operation principles of hydraulic turbines and pumps. This topic is usually introduced to mechanical engineering students in a course on applied fluid mechanics. Details of the subject matter can be found in most fluid mechanics textbooks (e.g. [8–11]). An important category of hydraulic turbomachines is the hydraulic reaction turbine. In addressing hydraulic reaction turbine problems, beside computing velocities,

losses and other variables, the main aim is to study the effects of varying certain parameters on the performance of the machine and to investigate conditions that minimize losses and maximize power and efficiency. Owing to the extensive computations involved, hand-calculations are not feasible. This forces the student to investigate, at best, the effects on machine performance of varying some selected variables and to accept without proof the effects of varying the remaining parameters. The majority of engineering students now have access to a personal computer and can benefit from the advances in this technology (speed, graphical capabilities, etc.) to solve the above time-consuming problem.

To this end, this paper describes REACTION, a microcomputer based, interactive, and menu-driven software package developed at the American University of Beirut for use in exploring the effects of design changes on the performance of hydraulic reaction turbines. REACTION relieves the mechanical engineering student from the monotonous task of performing repeated hand-calculations, it is much easier to use than a spreadsheet, which is still time-consuming as it requires development of all equations and formatting of graphics.

THE HYDRAULIC REACTION TURBINE

A hydraulic reaction turbine is one in which a water pressure drop occurs through both the guide vanes and the rotating runner vanes. The two types of reaction turbines in general use are the Francis turbine and the propeller turbine. The flow through these types is classified with respect to the flow line direction at entrance and discharge edges of the runner. For the Francis type, the design may be purely radial flow, where both entrance and discharge edges of the runner vanes are parallel to the axis of rotation (Fig. 1(a)); radial flow, where the discharge velocity has both radial and axial components (Fig. 1(b)); or mixed flow, where flow lines have radial and axial components throughout the runner vanes (Fig. 1(c)). For the propeller type, both entrance and discharge edges of the runner vanes are perpendicular to the axis of rotation (Fig. 1(d)). The transformation of potential energy into useful work in a hydroelectric, reaction turbine-based, power plant is described next.

After exiting the penstock (Fig. 2(a)), water enters a scroll or a spiral case (Fig. 2(b)) and then moves through a series of guide vanes where the flow is accelerated and given a definite tangential velocity component. For large turbines, stay vanes exist between the entry to the scroll case and the guide vanes; their main function is to support the weight of the generator above the runner. Water velocity also increases in flowing through these vanes because of their decreasing cross-sectional area. After leaving the runner, water enters a draft tube (Fig. 2(a)) which has an increasing cross-sectional area so as to reduce the flow velocity, and hence the head loss at submerged discharge.

For a reaction turbine the net head is defined as the difference between the total head at entrance to the scroll case and that at the tailrace (Fig. 2(a))

$$h = H_B - H_C \quad (1)$$

The head effectively available at the runner, h' , is the difference between the net head and the head losses in the draft tube and at submerged discharge. The head h'' that is extracted from water and transmitted to the runner is the difference between h' and head losses in the scroll case, guide vanes, and runner vanes. The extracted power $\gamma Q h''$ is further degraded by bearing friction and air resistance losses to yield shaft power.

Applying the angular momentum equation for a control volume enclosing the runner, the torque input to the shaft [8] is found to be

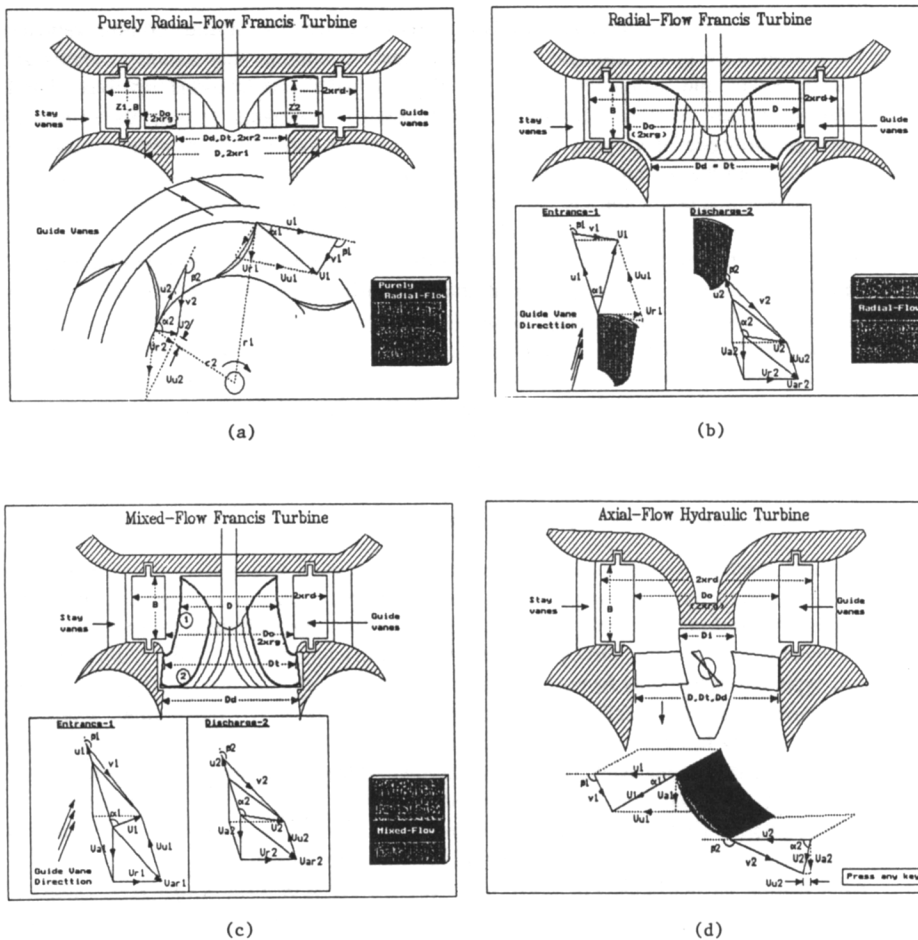


Fig. 1. Schematic of hydraulic reaction turbines.

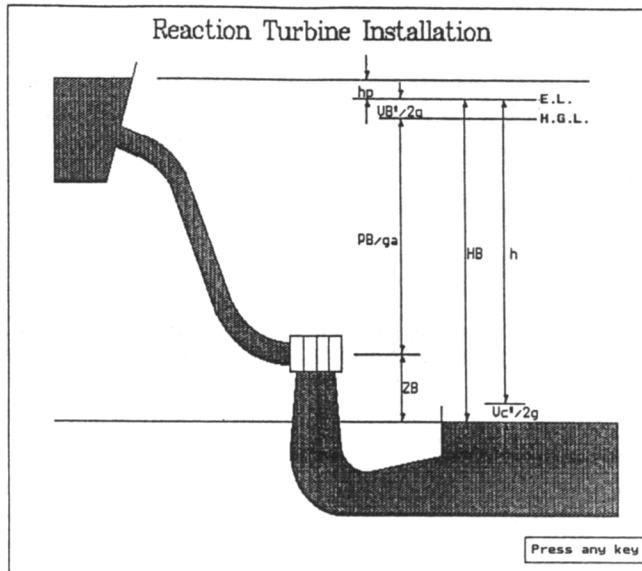
$$T = \rho Q (r_1 V_1 \cos \alpha_1 - r_2 V_2 \cos \alpha_2) \quad (2)$$

where the meanings of the terms in the above equation and the equations to follow are as indicated in the Nomenclature. Using the above equation, corresponding equations can be derived for the tangential force, hydraulic and overall efficiencies, output power etc., and are not presented here for compactness.

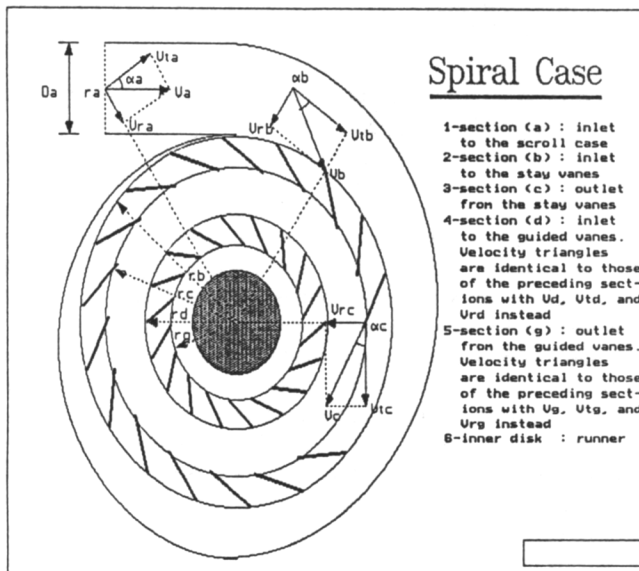
The specific speed of the reaction turbine is defined as [8]:

$$n_s = \frac{rpm_c \sqrt{bp}}{h^{5/4}} \quad (3)$$

As given by the above equation, the specific speed is not a dimensionless quantity. However, units are omitted in this paper in conformity with adopted practices in the literature. For Francis runners, when using the SI system of units, specific speeds range from 40 to 400; for



(a)



(b)

Fig. 2. (a) Reaction turbine installation; (b) scroll case assembly.

propeller runners, where high power output is obtained under low heads, the range is from 400 to 1000.

Finally, cavitation occurs in reaction turbines whenever the local absolute pressure drops to the water vapour pressure at the given temperature. This results in pitting, mechanical vibration, and low efficiencies. The critical factor in predicting cavitation is the vertical distance from the runner to the tailwater z_B (Fig. 2(a)), and it is used to define [8] a cavitation parameter σ with the following equation

$$\sigma = \frac{p_{\text{atm}}/\gamma - p_v/\gamma - z_B}{h} \quad (4)$$

The minimum or critical value of the cavitation parameter σ_C [8], which is determined experimentally as a function of the specific speed, sets a maximum limit for the elevation z_B given by

$$z_{B\text{max}} = p_{\text{atm}}/\gamma - p_v/\gamma - h\sigma_C \quad (5)$$

For complete details of the subject matter and of the equations used in the program, the reader should consult reference [8].

DESCRIPTION OF THE PROGRAM

The program is written in the Pascal computer language using the Turbo Pascal compiler, version 6 [12], and it runs on IBM PCs, or compatibles. The PC environment was chosen to provide an easy-to-use and cost-effective workstation. The user-friendly, menu-driven structure of the program provides a powerful teaching aid.

The software is divided into three major modules. The first provides menus and data entry windows for variables and assumptions. These windows can also be used for separate data retrieval. A complete set of mostly used unit systems is available, and the user can input and view each variable in any system of units. The purpose of the second module is to solve for unknown variables in an iterative manner, using the appropriate set of equations. The role of the third module is to output results in tabular forms employing either the Imperial or the SI system of units. An important feature of the program is its ability to plot the variation, as a function of a chosen variable, of up to five quantities at a time. Finally, diagrams of the inlet and discharge velocities can be drawn if corresponding data are complete.

Other options and features are also included to facilitate its use and enhance its capabilities. The first option is a file-handling utility that allows saving, opening, printing, and deleting data files consisting of input and output data for problems. Using this option, easy correction of erroneous or missing data is permitted. Moreover, this facility allows the user to return variables to their initial or previous values. Another feature of the program consists of three recommended charts [8] (Figs 3(a), (b) and (c)) showing the variation of some variables versus others. The user can retrieve values from the charts, and data can be used directly in solving problems requiring some initial assumptions and guesses. Also included in the package are graphical utilities consisting of six diagrammatic sketches of the reaction turbine [8]. The first (Fig. 2(a)) shows the turbine installation. The following four sketches (Figs 1(a), (b), (c) and (d)) illustrate the various types of reaction turbines, and the final one (Fig. 2(b)) displays the spiral case assembly. These sketches help clarifying and visualizing some variables used in the software (dimensions, head losses, velocity triangles). Furthermore, the program is equipped with an *on-line* help, for all available options, so as to

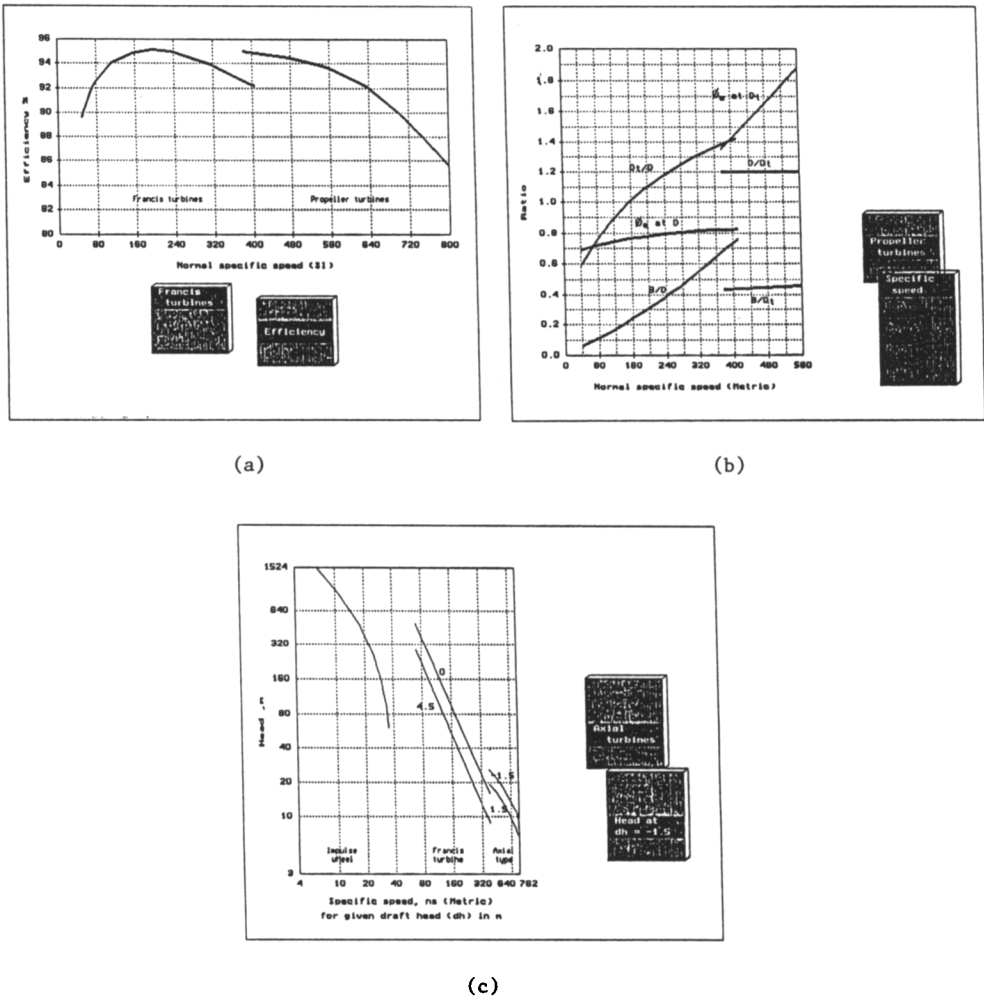


Fig. 3. (a) Recommended optimum values of turbine efficiency at a given specific speed; (b) recommended turbine dimensions and characteristics at a given specific speed; (c) recommended limits of specific speed for hydraulic turbines under various effective heads.

guide the user and to permit his learning about the utilities of the software easily. Finally, errors in any input operation, errors that will stop the execution of the program, and illegal input values for variables (negative dimension or loss factor greater than one) are carefully prevented.

The hierarchical structure of the software is shown in Fig. 4. The program is menu-driven and every interaction with the user is done in a user-friendly environment. The execution starts by typing REACTION at the DOS prompt, which causes the small menu to be displayed leading to the main menu which offers eight entries (Figs 4 and 5). By selecting the first entry (HELP), the user can access some information and instructions related to the software. The second entry (FILE) permits loading previously saved problems in addition

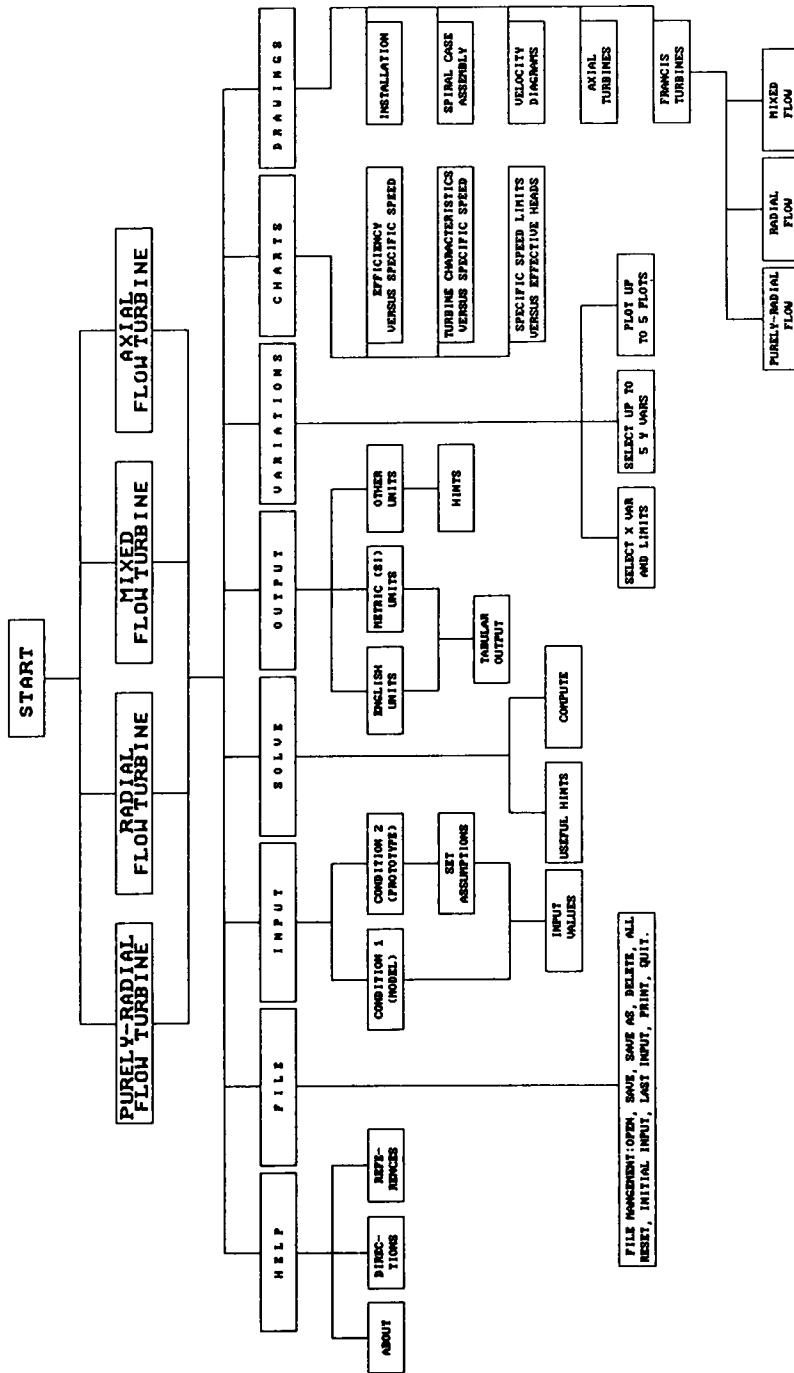
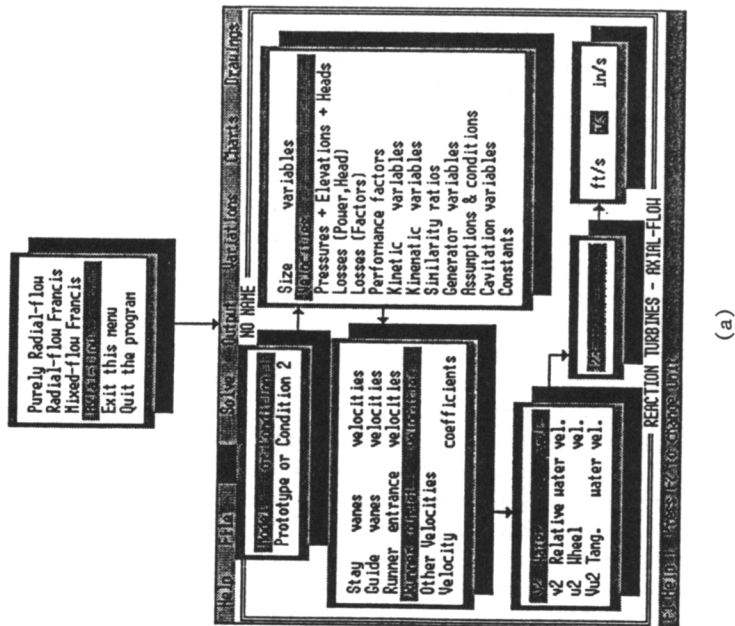
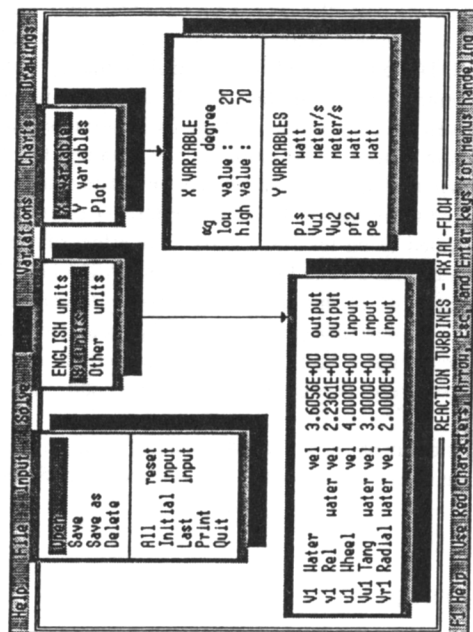


Fig. 4. Hierarchical menu structure of the package.



(a)



(b)

Fig. 5. Main menu and several illustrative sub-menus of REACTION.

to saving, printing, and deleting data files (Fig. 5(b), left-hand side). To start a new problem, the INPUT entry should be chosen. Here the student may select sub-entry MODEL or PROTOTYPE. Entry PROTOTYPE should be picked if similarity laws are to be invoked. Having decided on a sub-entry, known variables may be entered to the program. As depicted in Fig. 5(a), the menu-driven structure of the package facilitates this task by guiding the user throughout the data-entering procedure. Solution is then obtained by choosing the SOLVE entry. At this and the previous stage, data can be saved in a file that can be read at any later time. Results can be viewed using the OUTPUT or the INPUT entries (Fig. 5(b), lower left-hand side) and can be printed using the FILE entry. The effect of varying a parameter on other parameters is obtained, as shown in Fig. 5(b), in the VARIATIONS entry. This facility allows a comprehensive analysis of the different parameters involved in the problem and permits the optimization of the machine performance. The recommended charts (Fig. 3) and the diagrammatic sketches (Figs 1 and 2) can be accessed through the CHARTS and DRAWINGS entries, respectively. Using also the last command, velocity diagrams can be drawn.

If the IBM 'graphics' module has been loaded at the beginning of the session, a hard copy of figures displayed on the screen can be obtained using the 'Print Screen' command with a suitable printer.

EXAMPLE CALCULATIONS

In this section, an example problem demonstrating the computational and graphical capabilities of the program and how it leads to a deeper understanding of the subject material is presented. Records of the input and output data of the problem are listed in Table 1. The task is to select the outlet angle of the inlet guide vanes (α_g), the decision being based on optimizing some performance parameters such as power input to shaft (pis), net head (h), and hydraulic efficiency (η_h). The variation of the power input to shaft is depicted in Fig. 6(a). To generate this or any other plot, the user should select the x and y variables by choosing the VARIATIONS entry from the main menu (Figs 4 and 5(a)). Having done this, the user will be asked to specify, in simple dialog boxes, values for the low and high limits of the x variable, y variable(s) (maximum number of five), and the corresponding pair of system units. The resulting specification will be shown in a window similar to the one on the right-hand side of Fig. 5(b). Then, by choosing the plot command, the x interval will be subdivided into 45 sub-intervals and the problem will be solved 45 times in order to compute and store y values for the graphical plots.

As shown in Fig. 6(a), the power input to shaft (pis) maximizes for a definite value of α_g . To understand this behaviour, the plots of Figs 6 and 7 and the following explanation are needed. Since the mass flow rate is constant, the radial velocity of water at exit from the guide vanes (V_{r_g}) should be constant. Then, as α_g increases, the tangential water velocity at exit from the guide vanes (V_{t_g}) will decrease (refer to the velocity triangle in Fig. 2(b)). Since by neglecting losses the moment of momentum in the space between the guide vanes and rotor inlet is conserved, the tangential water velocity at rotor inlet (V_{u_1}) should decrease, as depicted in Fig. 6(b). Since the losses in the stay and guide vanes are neglected, the total head at inlet (H_1) will be equal to the total head at inlet to the casing (H_B) and will remain constant at constant mass flow rate. Therefore, the decrease in V_{u_1} at constant head (H_1) and rotational speed (rpm) will lead to a decrease in the tangential water velocity at exit (V_{u_2}) (Fig. 6(c)), work being done on the rotor. Now, with the aid of the velocity diagram at exist shown in Fig. 1(d), the decrease in V_{u_2} at constant runner velocity (u_2) and axial water

Table 1. Input and output data for an axial-flow reaction turbine problem. (Output is in SI units, angles in degrees, angular velocity in radian/s.)

Input Data—Model or Condition 1		Other velocities	Generator variables
<i>Ratios + linear dim. variables</i>		<i>n</i> RPM	Electricity frequency Hz
<i>x</i> # of turbines	: 1.000		: 60.000
r_1 Wheel radius at entrance	: 1.375		
r_2/r_1	: 1.000	<i>Pressures + elevations + heads</i>	
D_1, D, D_d Maximum diameter	: 4.000	P_2 Runner outlet pressure	Assumptions & conditions
D_1 Hub diameter	: 1.500	z_B Casing inlet elevation	Use efficiency vs n_s : NO
		z_1 Runner inlet elevation	Use turbine charac. vs n_s : NO
		z_2 Runner outlet elevation	
<i>Surface + angle variables</i>		H_B Casing inlet total head	<i>Cavitation variables</i>
m Inlet axial area coefficient	: 0.960		z_{Bmax} Max turbine runner elev. : 4.000
			Elevation above sea level : 500.000
			Temperature °C : 15.000
<i>Guide vanes variables</i>		<i>Losses (power, head)</i>	
α_g Angle between V_g & Vu_g	: 30.000	hf_1 Guide & stay vane losses	Constants
r_d Guide Vane inlet radius	: 3.950	hf_3 Draft tube losses	g acc. of gravity : 9.815
D_o Guide vane outlet diameter	: 6.500	h_s Submerged discharge loss	Fluid density : 998.753
B Guide vane height	: 1.500	<i>Losses (factors)</i>	Fluid specific weight : 9802.259
m_d Guide vane in. area coeff.	: 0.965	K Runner vanes loss coefficient	
m_g Guide vane out. area coeff.	: 0.960		
<i>Stay vanes variables</i>		<i>Performance factors</i>	Output Data—Model or Condition 1
r_a Scroll case entrance radius	: 7.000	n_v Volumetric efficiency	<i>Ratios + linear dim. variables</i>
r_b Stay vane inlet radius	: 5.100	n_m Mechanical efficiency	r_2 wheel radius at discharge : 1.375
r_c Stay vane outlet radius	: 4.500		D_{od} Draft outlet diameter : 3.065
B_b Stay vane inlet height	: 2.000	<i>Kinematic variables</i>	B/D_1 : 0.375
B_c Stay vane outlet height	: 1.750	Q Initial discharge/turbine	D_o/D_1 : 1.625
D_a Scroll case inlet dia.	: 2.750		
m_b Stay vane inlet area coeff.	: 0.975	<i>Similarity ratios</i>	<i>Surface + angle variables</i>
m_c Stay vane outlet area coeff	: 0.970	D_1/D_2	α_1 Angle between V_1 & u_1 : 33.903
		RPM_1/RPM_2	α_2 Angle between V_2 & u_2 : 56.655

Table 1. (continued)

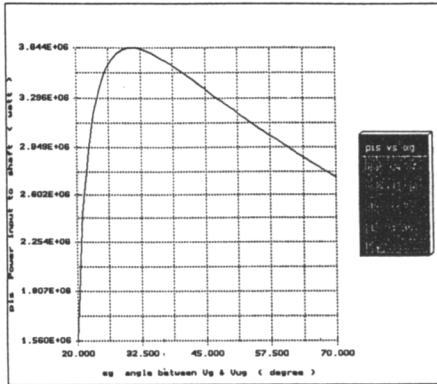
<i>Losses (factors)</i>			Input Data—Prototype or Condition 2	
K' Draft tube loss coefficient-1	: 0.000		<i>Similarity ratios</i>	
K_d Draft tube loss coefficient-2	: 0.000		D_1/D_2	: 0.800
			RPM_1/RPM_2	: 0.500
<i>Performance factors</i>			<i>Assumptions & conditions</i>	
n_s Specific speed	: 387.480		Use efficiency vs n_s	: NO
η_h Hydraulic efficiency	: 0.747		Use turbine charac. vs n_s	: NO
n_o Overall efficiency	: 0.695			
ϕ Peripheral vel. factor at r_1	: 1.092			
ϕ_1 Peripheral tip vel. factor	: 1.589			
<i>Kinetic variables</i>			<i>Constants</i>	
p_w Inlet water power at casing	: 6819161.283		g acc. of gravity	: 9.815
pir Power input to runner	: 6614586.444		Fluid density	: 998.753
pis Power input to shaft	: 3642485.331		Fluid specific weight	: 9802.259
pb Brake power per turbine	: 3496785.918			
pb_1 Installation brake power	: 3496785.918			
pe Exit water power from runner				
	: 1736254.824			
T Torque input to shaft/turbine	: 347995.492			
T_f Friction torque/turb.	: 13919.820			
T_s Out, shaft torque/turb.	: 334075.672			
<i>Kinematic variables</i>			Output Data—Prototype of Condition 2	
Q_1 Leakage discharge/turbine	: 1.740		<i>Ratios + Linear dim. variables</i>	
Q_e Effective discharge/turbine	: 56.260		D_t, D, D_d Maximum diameter	: 5.000
Q_t Total discharge	: 58.000		B/D_t	: 0.375
h Net head	: 8.850		D_o/D_t	: 1.625
			<i>Surface + Angle variables</i>	
			m Inlet axial area coefficient	: 0.960
			<i>Guide Vanes variables</i>	
			r_g Guide vane outlet radius	: 4.063
			D_o Guide vane outlet diameter	: 8.125
			B/D_o	: 0.231

Table 1. (continued)

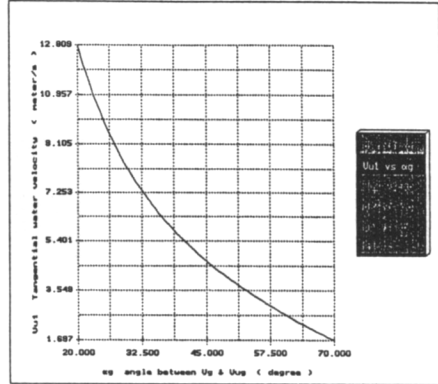
β_1 Angle between v_1 & u_1	: 139.366	V_{1b} Stay vanes inlet tang. vel.	: 2.177	<i>Other velocities</i>	
β_2 Angle between v_2 & u_2	: 153.381	V_{1b} Stay vanes inlet radial vel.	: 0.928	V_d Inlet draft tube velocity	: 4.615
a_1 Wheel in. area normal to v_1	: 6.751	V_c Stay vanes outlet vel.	: 2.747	V_3 Submerged discharge vel.	: 7.863
a_2 Wheel out. area normal to v_2	: 4.645	V_{1c} Stay vanes outlet tang. vel.	: 2.467	V_c Tailrace velocity	: 7.863
A_1 Wheel in. area normal to V_1	: 5.783	V_{1c} Stay vanes outlet radial vel.	: 1.208	w Rotative velocity	: 10.472
A_c Wheel inlet axial area	: 10.367				
<i>Guide vanes variables</i>					
α_d Angle between V_d & V_{ud}	: 29.871	<i>Guide vanes velocities</i>		<i>Velocity coefficients</i>	
r_g Guide vane outlet radius	: 3.250	V_d Guide vanes inlet vel.	: 3.242	C_1 V_1 Velocity coefficient	: 0.738
B/D_o	: 0.231	V_{1d} Guide vanes inlet tang. vel.	: 2.811	C_2 V_2 Velocity coefficient	: 0.493
A_d Guide vane in. radial area	: 35.925	V_{1d} Guide vanes in. radial vel.	: 1.614	C_r V_r Velocity coefficient	: 0.150
A_g Guide vane out. radial area	: 29.405	V_g Guide vanes outlet vel.	: 3.945		
		V_{1g} Guide vanes outlet tang. vel.	: 3.416	<i>Pressures + elevations + heads</i>	
		V_{1g} Guide vanes out. radial vel.	: 1.972	P_B Casing inlet pressure	: 30812.580
<i>Stay vanes variables</i>					
α_a Angle between V_a & V_{ua}	: 80.652	<i>Runner entrance velocities</i>		P_1 Runner inlet pressure	: 52245.878
α_b Angle between V_b & V_{ub}	: 23.091	V_1 Water velocity	: 9.729	H_1 Runner inlet total head	: 12.000
α_c Angle between V_c & V_{uc}	: 29.094	v_1 Relative water velocity	: 8.333	H_2 Runner outlet total head	: 3.150
A_a Scroll case inlet area	: 5.940	u_1 Wheel velocity	: 14.399	H_c Tailrace velocity head	: 3.150
A_b Stay vane inlet radial area	: 62.486	V_{u1} Tangential water velocity	: 8.075		
A_c Stay vane outlet radial area	: 47.996	V_a Axial water velocity	: 5.427	<i>Losses (power, head)</i>	
		u_1 Wheel velocity at dia. D_1	: 20.944	hf_2 Runner losses	: 2.242
				H_1 Tailrace (H_c) loss	: 3.150
<i>Stay vanes velocities</i>					
V_a Scroll case inlet (V_B) vel.	: 9.765	<i>Runner outlet velocities</i>		pf_1 Guide & stay vanes power loss	: 0.000
V_{1a} Scroll case inlet tang. vel.	: 1.586	V_2 Water velocity	: 6.496	pf_2 Runner (hydraulic) power loss	: 1235846.289
V_{1a} Scroll case inlet radial vel.	: 9.635	v_2 Relative water velocity	: 12.112	pf_3 Draft tube power loss	: 0.000
V_b Stay vanes inlet vel.	: 2.367	u_2 Wheel velocity	: 14.399	p_s Submerged discharge power loss	: 0.000
		V_{u2} Tangential water velocity	: 3.571	p_t Tailrace power loss	: 1789953.427
				pmf Mech. power loss	: 145699.413
				pl Leakage power loss	: 150876.236

Table 1. (continued)

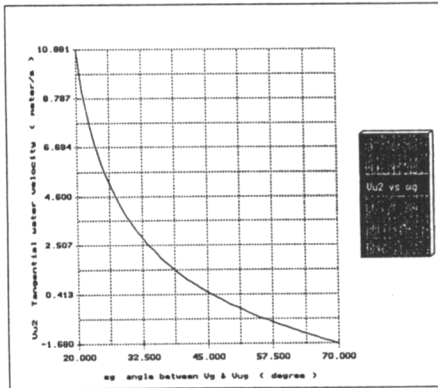
		<i>Cavitation variables</i>	
		σ_c	Critical cavit. param. (min)
<i>Guide vane heights</i>			
B	Guide vane height	1.875	0.970
A_g	Guide vane out. radial area	45.946	0.747
m_d	Guide vane inlet area coeff.	0.965	0.978
m_g	Guide vane outlet area coeff.	0.960	0.709
<i>Stay vane variables</i>			
m_b	Stay vane inlet area coeff	0.975	1.589
m_c	Stay vane outlet area coeff	0.970	
<i>Guide vane velocities</i>			
V_{r_g}	Guide vanes out. radial vel.	4.931	
<i>Runner entrance velocities</i>			
u_t	Wheel velocity at dia. D_t	52.360	
<i>Other velocities</i>			
V_d	Inlet draft tube velocity	11.539	
w	Rotative velocity	20.944	
n	<i>RPM</i>	200.000	
<i>Velocity coefficients</i>			
C_r	V_{r_g} Velocity coefficient	0.150	
<i>Losses (power, head)</i>			
pmf	Mech. power loss	1924146.151	
<i>Performance factors</i>			
n_s	Specific speed	391.168	
<i>Kinematic variables</i>			
Q	Initial discharge/turbine	226.562	
Q_l	Leakage discharge/turbine	6.797	
Q_c	Effective discharge/turbine	219.766	
h	Net head§	55.313	
h''	Head extracted from water	41.301	
<i>Similarity ratios</i>			
h_1/h_2		0.160	
Q_1/Q_2	per wheel	0.256	
T_1/T_2	Torque in to shaft/wheel	0.082	
pis_1/pis_2	Power in to shaft/wheel	0.041	
Ts_1/Ts_2	Out shaft torque/wheel	0.080	
pb_1/pb_2	Brake power/wheel	0.040	



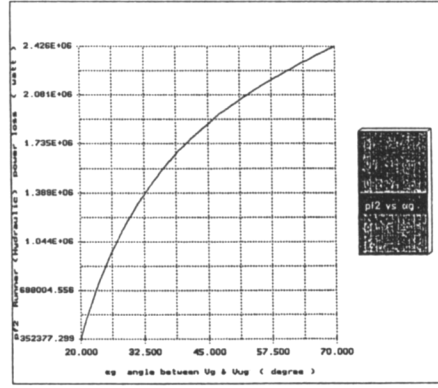
(a)



(b)



(c)



(d)

Fig. 6. Variation with guide vanes direction angle of (a) power carried by water exiting the turbine runner, (b) net head, and (c) hydraulic efficiency angle.

velocity (V_{a2}) (since the flow rate remains unchanged) will first result in an increase in the relative exit water velocity (v_2) and a decrease in the absolute exit water velocity (V_2) until Vu_2 changes sign. When Vu_2 becomes negative, v_2 will still increase but, on the other hand, V_2 will start increasing. Since the hydraulic power loss (pf_2), or rotor friction loss, is proportional to v_2 , and the exit water power (pe) is proportional to V_2 when no draft tube is used, the behaviour of the curves in Figs 6(d) and 7(a) is understood. Moreover, because the power input to the runner is constant and equal to the sum of pis , pf_2 , and pe , it should be apparent that the increase in pf_2 and the decrease in pe will result in a value of α_g that will minimize their sum and maximize pis (Fig. 6(a)).

The effect of varying α_g on the net head (h) is as depicted in Fig. 7(b). This behaviour is expected because in this problem, where no draft tube is employed, h is the difference between the head at inlet to the casing H_B and the total head at rotor exit (H_2), which is proportional to pe at constant mass flow rate; therefore, H_B being constant, h and pe should

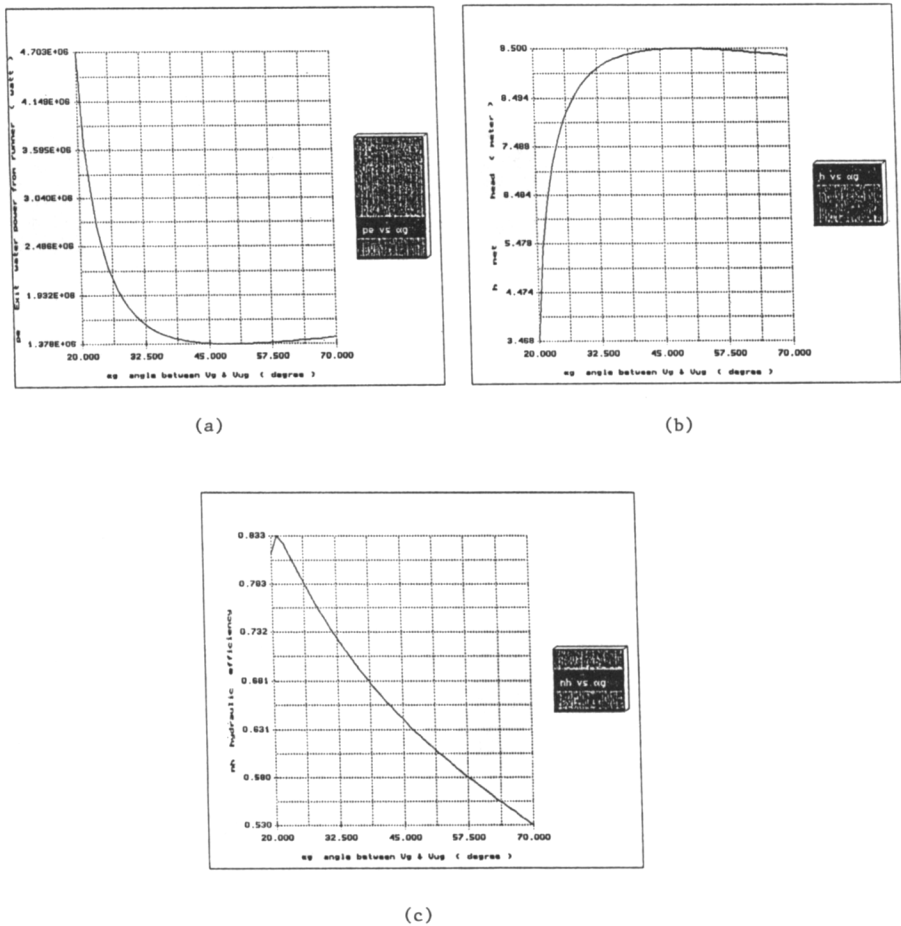


Fig. 7. Variation with guide vanes direction angle of (a) power carried by water exiting the turbine runner, (b) net head, and (c) hydraulic efficiency angle.

vary in an opposite way (compare curves in Figs 7(a) and (b)). Finally, since the hydraulic efficiency (η_h) is defined as the ratio of the head extracted from water (h'') to the net head h , and since h'' is proportional to P_{out} , η_h will first increase with the sharp increase in h'' and then decrease as h'' decreases and h remains approximately constant (Fig. 7(c)). This is to be anticipated since the overall effect of increasing α_g is to continuously increase the friction losses in the runner and decrease the losses at exit to a value approximately constant over most of the range of α_g . Using the value of α_g that maximizes P_{out} , say 30 degrees, the problem is solved, and the values of the various quantities are displayed in Table 1. The velocity triangles at inlet and exit from the runner are shown in Figs 8(a) and (b), respectively. Moreover, the variations of the blade angles at inlet (β_1) and exit (β_2) from the runner with r_1 are shown in Figs 9(a) and (b) respectively. The values obtained are to be used when the runner is to be designed under these flow conditions. Finally, doubling the rotational speed and increasing the diameter by a factor of 1.25, the performance of the turbine is easily obtained using similarity laws (Table 1).

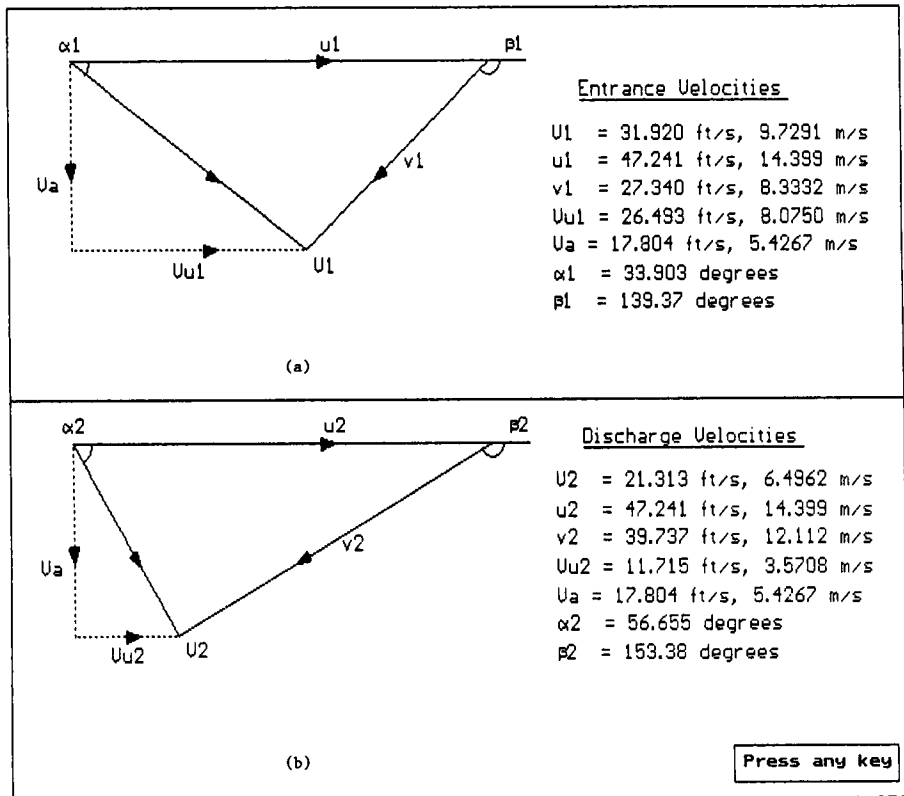


Fig. 8. Velocity triangle at runner (a) inlet and (b) exit.

CONCLUSION

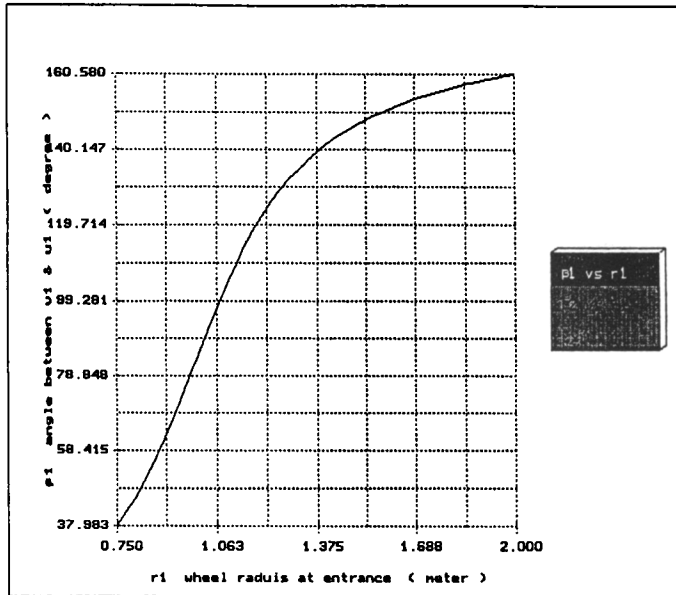
An educational software package for the analysis of hydraulic reaction turbines was described. The program, capable of solving radial, mixed, and axial flow hydraulic reaction turbine problems, provides an efficient educational tool for mechanical engineering students. The comprehensive example problem presented demonstrated the various features of the package. Finally, copies of REACTION will be provided to users upon request.

ACKNOWLEDGEMENT

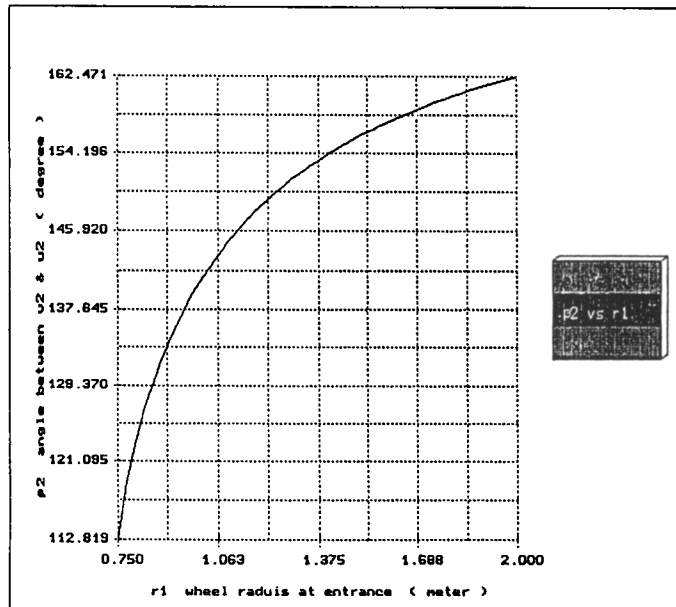
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(a)



(b)

Fig. 9. Variation of blade angle at runner (a) inlet and (b) exit with wheel radius.

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