MODELING OF OFF-DESIGN MULTISTAGE TURBINE PRESSURES BY STODOLA'S ELLIPSE

By:

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Presented at:

ENERGY INCORPORATED

PEPSE® USER'S GROUP MEETING

RICHMOND, VIRGINIA

NOVEMBER 2-3, 1983

BECHTEL POWER CORPORATION

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ABSTRACT

The conventional method for predicting the variation of multistage turbine bowl and shell pressures with flow in many commercially available heat and mass balance computer programs, including PEPSE, is that of Constant Flow Coefficient. This method produces reasonably accurate results, and has long been used for flow through many consecutive stages and casings arranged in series with uncontrolled extractions, exhausting to vacuum, as is common in the utility power industry.

For controlled extraction and high-back-pressure turbine designs common in process utility and cogeneration plants, where the pressure at one or more points in the expansion is held constant, by varying the flow area, the Constant Flow Coefficient method is not valid. The paper presents schematic examples and brief descriptions of cogeneration designs, with background and derivation of a more generalized "nozzle analogy" method which is applicable in these cases. This method is known as the Law of the Ellipse. It was originally developed experimentally by Professor Stodola and published in English in 1927. The paper shows that the Constant Flow Coefficient method is really a special case of the more generalized Law of the Ellipse. The complete equation not only allows pressure prediction for controlled extractions and high back-pressure, but also provides a more consistent and well-behaved method for the last few low-pressure stages in high-vacuum, series-flow arrangements.

Graphic interpretation of the Law of the Ellipse for controlled and uncontrolled extractions, and variations for sonic choking and reduced number of stages (including single stage) are presented. The derived relations are given in computer codable form, and methods of solution integral with overall iteration schemes are suggested, with successful practical experience. In addition to modeling of pressures, the rudiments of a dynamic similarity method of modeling turbine efficiencies are also presented.

INTRODUCTION

Pressure-flow relations in a cascading multistage turbine, where the intermediate expansion pressures may vary with nozzle flow coefficients and blade flow angles as the reaction changes at off-design loads, are very complex phenomena. Usually, the turbine manufacturer provides curves of key pressures vs. flow for normal operating load changes, and heat balances which establish pressure and flow at all significant points based on detailed row-by-row calculations which take account of such complex phenomena. These heat balances are usually provided at "maximum calculated", 100, 75, 50 and 25 percent of normal design or guaranteed output and are generally regarded as the closest thing to true operating conditions unless or until modification to conform to actual test measurement takes place.

There are, of course, many times during design of a power plant and after commencement of commercial operation, when an owner or engineer may need to explore off-design and/or abnormal conditions between or beyond the normal load points provided by the manufacturer. Heaters out of service or degraded in performance, emergency by-pass of turbine generators after a load-trip, capacity of auxiliary systems to handle emergency heat loads due to seal failure, steady state extremes for transient system analyses and economic effect of possible cycle modification alternatives are but a few occasions where accurate, exploratory heat balance analysis is necessary. PEPSE and other similar programs have greatly benefitted the utility power industry by providing the means to accomplish this work expeditiously and elegantly.

When available, the basic "thermal kit" provided by the manufacturer, containing the load-point heat balances and curves, are invaluable. Using these, the owner or engineer can interpolate and/or extrapolate from the basic load points to determine with reasonable accuracy the two essential unknowns for off-design performance: (1) bowl and shell pressures, and (2) turbine efficiency. To accomplish this, PEPSE and a few other high-quality programs provide consistent but limited sets of rules, the most common of which is that of Reference (1).

For early, conceptual cycle design work it is possible, using programs similar to PEPSE and available techniques in the literature, to approximate the cycle pressures of any turbine manufacturer to within ten percent and achievable efficiencies to within a few overall percentage points. This has been accomplished recently by the author's company in establishing a basic cycle of quite unique and advanced design, prior to bid of the turbine-generator. Working in this manner, it was possible to "standardize" the cycle, eliminating uneconomical alternatives which would have greatly complicated the bid evaluation, while at the same time providing sufficient latitude to allow three major manufacturers of different turbine design types to meet specification requirements to best advantage.

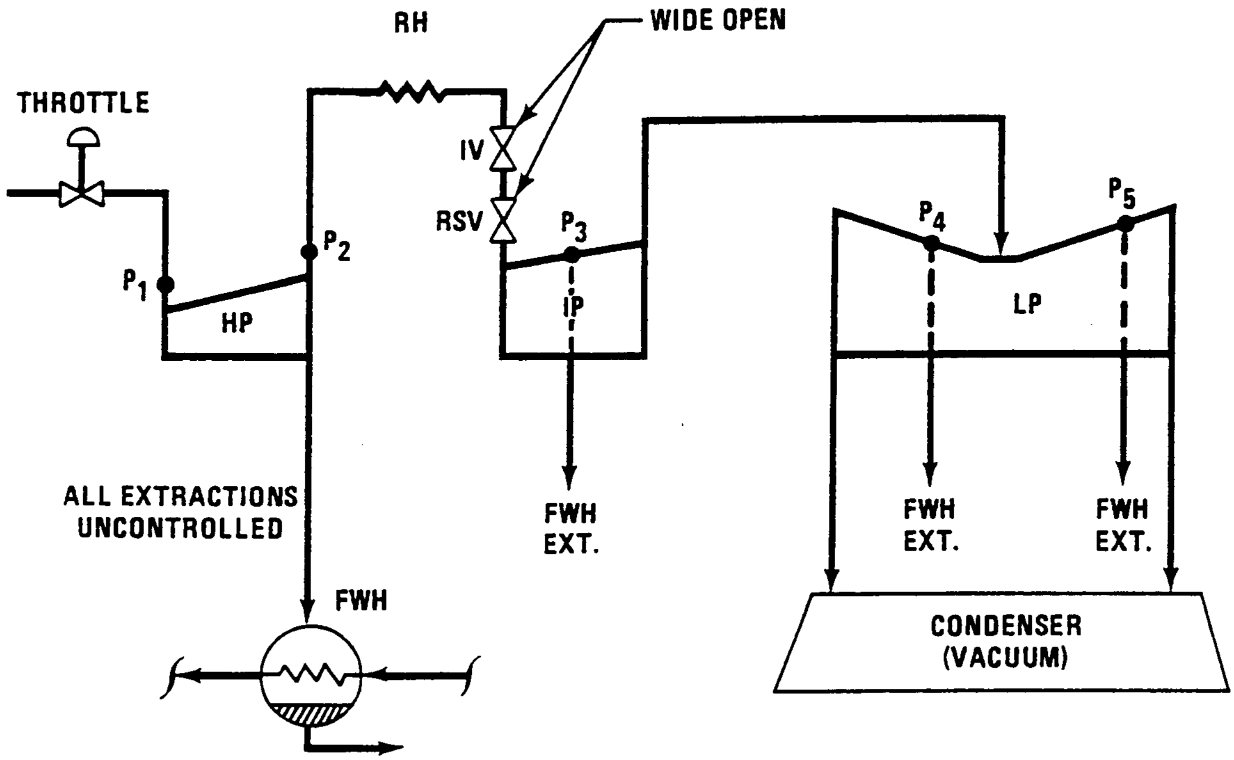
The purpose herein is to describe and derive one of the techniques used in the above described work, that of determining bowl and shell pressures for a controlled expansion, in contrast to the uncontrolled expansion which is typical of most utility power systems. By utilizing this technique, however, one area of uncontrolled expansion pressure modeling which could be improved, that of the last few LP stages, is pointed out.

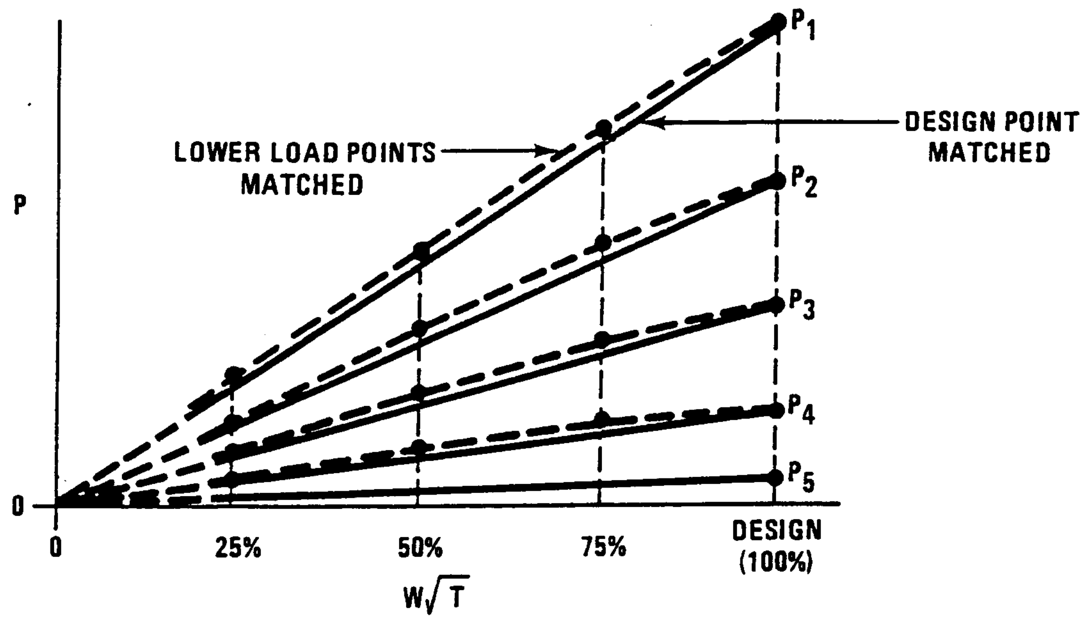
UNCONTROLLED VS. CONTROLLED EXPANSION

A schematic example of the typical utility power uncontrolled expansion to high vacuum is shown in Figure 1. Downstream of the throttle, the entire expansion is characterized by fixed flow areas, inasmuch as the reheat stop valve (RSV) and intercept valve (IV) are designed to remain fully open during all normal operating modes. This is in contrast to the throttle, which is designed to modulate at all loads below absolute maximum throttle flow, in order (in U.S. systems) to maintain constant pressure at boiler outlet. In U.S. utility practise the schematically shown throttle "valve" is, in actuality, a sequential valve steam chest with a variable admission governing stage. (The term "governing" applies when such a device is the primary control element in the

Figure 1

UNCONTROLLED MULTISTAGE EXPANSION
TO HIGH VACUUM





expansion, used for grid frequency response. Similar arrangements for other purposes downstream in the cycle are termed "control" stages elsewhere in this paper.)

In many European utility power systems (Benson design) there is no primary pressure control element. The pressure of the entire steam generator is varied by controlling firing rate, primarily. In such cases the "fixed flow areas" might extend all the way upstream through the boiler to the point of feedwater flow control, whether by throttling or variable speed. The relations described in this paper would not, however, extend into the boiling regime.

For the system shown in Figure 1 it has long been recognized, References (2) and (3), that at any point in the expansion downstream of the throttle the pressure-flow relation may be approximated by

$$\text{Mass Flow Coefficient} = \frac{W_i}{\sqrt{\frac{P_i}{v_i}}} = \text{Constant}, \qquad \text{(A)}$$

where,

W_i = Flow to next stage group, lbs/hr

P_i = Bowl or shell pressure, psia

v_i = Specific volume, ft^3/lb, corresponding to state point at P_i

i = Subscript denoting any point in the expansion.

Another expression for Mass Flow Coefficient is widely used in the literature where the perfect gas law, $Pv = RT$, can be assumed to apply. This form, Reference 4, is

$$\text{Mass Flow Coefficient} = \frac{W_i \sqrt{T_i}}{P_i} = \text{Constant}, \qquad \text{(B)}$$

where:

T_i = Absolute temperature, R, corresponding to the state point at P_i.

Although equation (A) is the more correct form based on turbomachinery theory, as a later appendix to this paper will show, it gives the misleading impression that absolute pressure is proportional to the <u>square</u> of mass flow. Equation (B) shows more clearly the nearly linear relationship between pressure and flow for an uncontrolled expansion to high vacuum. Regardless of fluid properties, either of equations (A) or (B) provides nearly the linear relationship shown by the solid lines in the lower part of Figure 1. The relative variation of absolute temperature at a given point in the expansion is usually so small that it can be neglected, and the relationship remains nearly linear in reasonable accord with practical data. The area most subject to question is in the very low flow regime where the relationship approaches zero pressure at zero flow.

Figure 1 indicates schematically that flow coefficient "schedules", available, with PEPSE and other programs, can be used to "match" the turbine manufacturer's lower load points, providing a segmented linear fit which more closely follows the row-by-row calculations.

Controlled Expansion - A simplified controlled expansion arrangement typical of a recent cogeneration application is shown in Figure 2. The expansion is broken up into several discrete segments, each preceded by variable flow area control devices represented as valves A, C, and F. The back pressures at the end of each segment, at points B, D and J, are held constant at high levels to meet the Owner's process steam requirements. These back pressures must remain constant at all flows, so the relations described in the preceding section cannot apply. As the process steam demand increases at point B, for example, valve C will automatically close to restore the pressure in the header. This reduces flow in the following segment. Theoretically, especially with regard to the pressure modeling relations, such increased demand at point B could proceed until the flow to the downstream segment reached zero. In practise, maximum limits are placed on process steam demand from the turbine at each controlled extraction, with emergency "let-down" sources, if necessary, to maintain header pressure, in order to avoid damage to downstream turbine and boiler components by reduced flow. In most cogeneration applications the maintenance of process steam at the required pressure and demand flow is the prime objective, and electric power generation becomes a variable by-product. However, sufficient boiler capacity may be designed to provide a suitable minimum power level, and steam process demands may be steady for long periods so that grid stability is not adversely affected.

Controlled extractions may be integral with the turbine casing as at point B, whence the turbine is known as a "single automatic extraction" machine. Double automatic extractions, with two controlled extractions in one casing, are common. Controlled extractions also occur between turbine casings as at point D, with control at E. The extraction and back-pressure control devices at C, E and F are most efficiently designed as sequential valve steam chests with variable admission control stages to reduce throttling, but the simpler and less expensive throttle valves shown in the figure are also commonly used.

Uncontrolled extractions for feedwater heating are usually included in the design as at G and H. The controlled back pressure segment between F and J is typical of pressure modeling on any uncontrolled (fixed flow area) segment in the cycle, and is therefore the example for theoretical derivations in the section to follow.

THE LAW OF THE ELLIPSE

Consider a multistage turbine expansion segment with several uncontrolled extraction groups as shown in Figure 3, where the final back pressure is some fixed value. For any extraction group, i, where i is 1, 2, or 3, as in the Figure, a "nozzle analogy", Reference (4), may be developed which treats each entire group expansion as if it were a single nozzle. This analogy is known as Stodola's Ellipse, References (4), (5) and (6), and states that

$$\Phi_i \propto \sqrt{1 - \left(\frac{B_i}{P_i}\right)^2} \qquad \text{(C)}$$

Figure 2

CONTROLLED EXPANSION COGENERATION SCHEMATIC
WITH HIGH BACK-PRESSURE

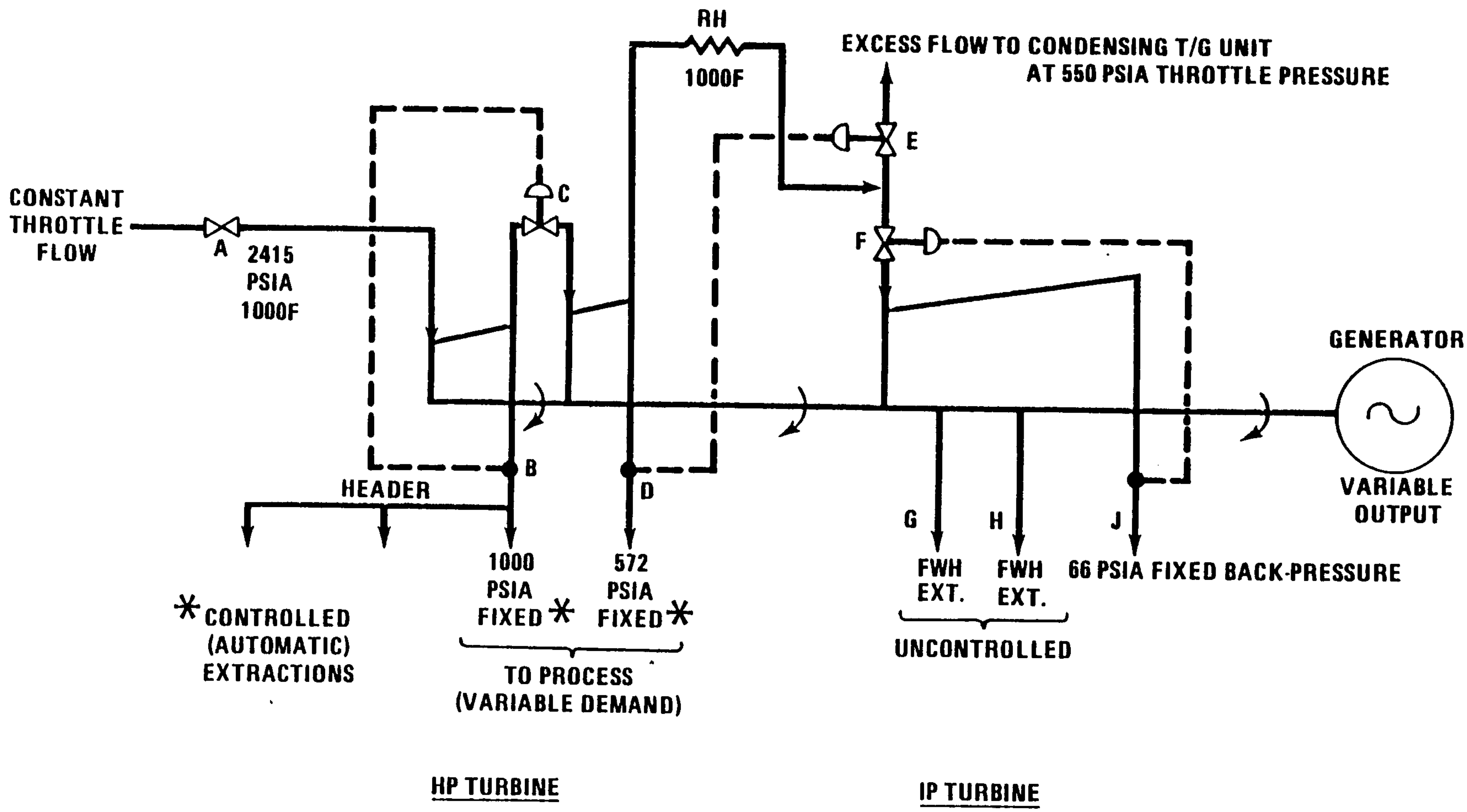
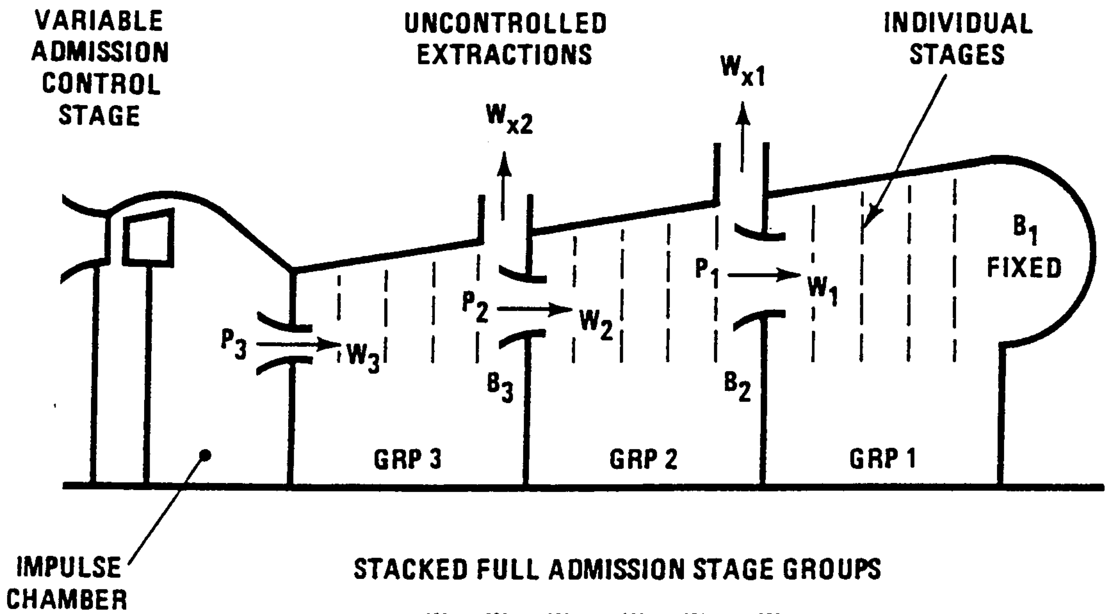


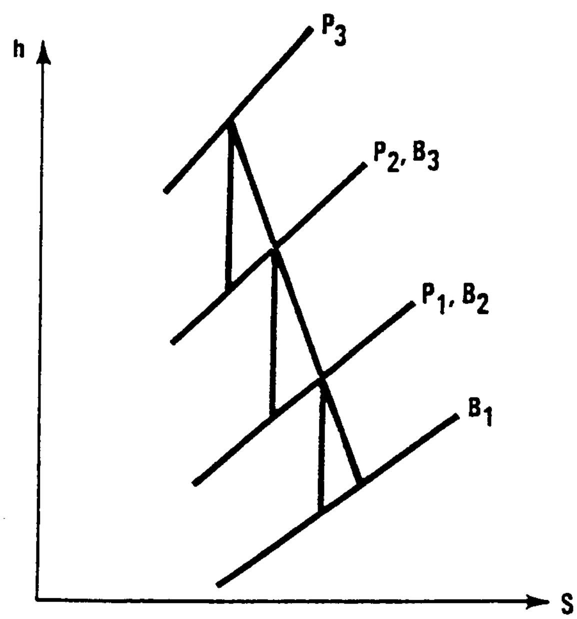
Figure 3

PRESSURE-FLOW ARRANGEMENT
FOR HIGH BACK-PRESSURE, FULL ADMISSION, MULTISTAGE SEGMENT



STACKED FULL ADMISSION STAGE GROUPS

$$W_3 = W_2 + W_{x2} = W_1 + W_{x1} + W_{x2}$$



where:

Φ_i = Mass Flow Coefficient, equation (A) or (B)

P_i = Inlet total pressure to the first stage nozzle of any group, psia

B_i = Exit static pressure from the last stage of any group, psia

This proportionality is the source of the Constant Flow Coefficient method since it will be noted that with a very low value of B_1, as with a high vacuum exhaust, the right hand term under the right radial is negligible and equation (A) or (B) results for all groups.

The proportionality in (C) was originally developed experimentally by Professor Stodola using an eight stage laboratory turbine at the Polytechnicum in Zurich early in this century. His "Cone of Steam Weights", showing the ellipses at various initial pressures, is presented in Figure 4. He states, Reference (5), that with suitably chosen scales the ellipses appear as circles as they do in the figure.

The development of the proportionally into the familiar elliptical equation with semimajor axis unity, is shown in Figure 5 from Reference (6). The proportionality holds down to some back-pressure B_t where one of the stage nozzles in the multistage group chokes due to sonic conditions. Thereafter the curve is flat. The choking point is usually much lower than the single nozzle choking value of about 0.5 because of the multiple stages.

Using velocity triangle and work vs. flow relations, it is possible to derive the elliptical proportionality from purely theoretical bases. A future appendix to this paper will show, from reference (6), that the resulting relation is

$$\frac{W_i}{\sqrt{\frac{P_i}{v_i}}} \propto \sqrt{1 - \left(\frac{B_i}{P_i}\right)^{\frac{n+1}{n}}} \qquad \text{(D)}$$

where:

n = polytropic efficiency,

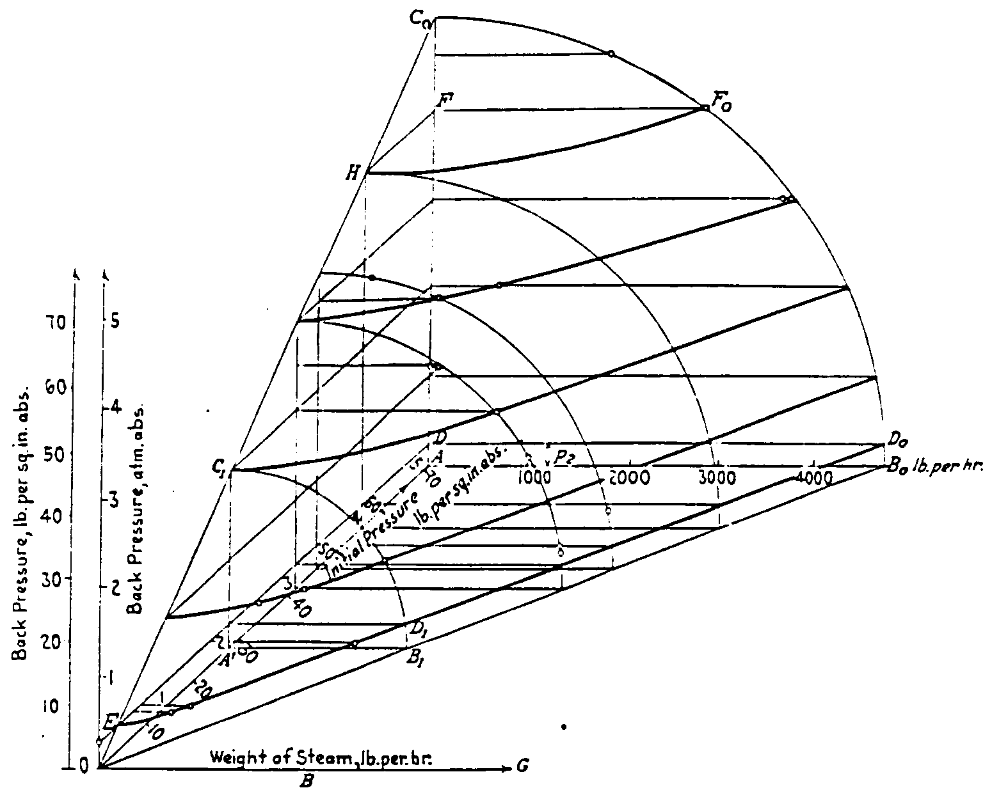
$$\frac{n-1}{n} = \eta_{st} \left(\frac{k-1}{k}\right) \qquad \text{(E)}$$

k = isentropic exponent

η_{st} = "small stage" efficiency, related to the slope of the expansion line on a Mollier diagram

Figure 4

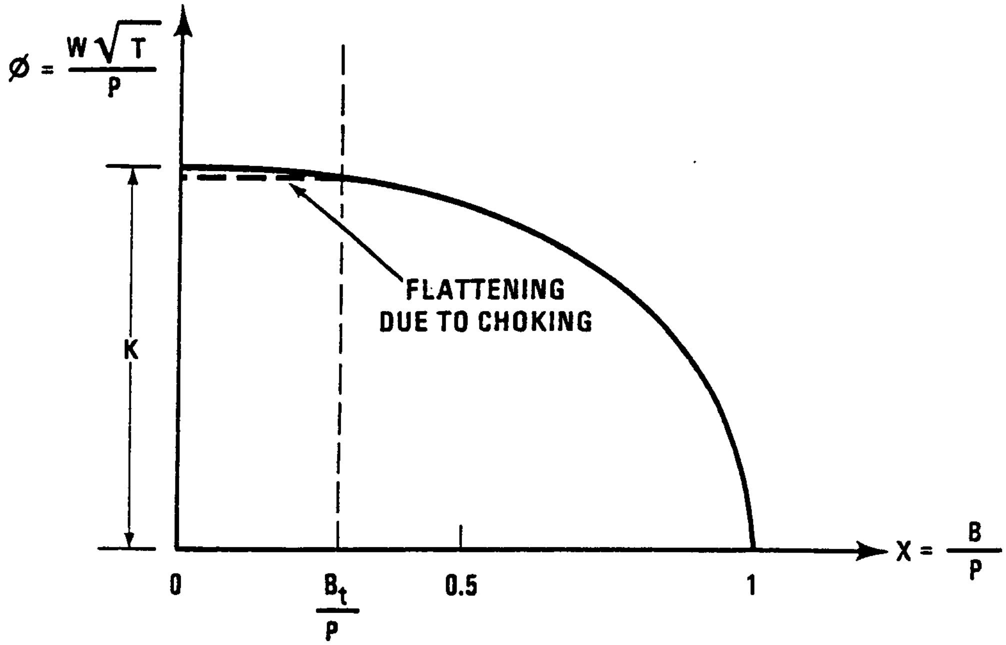
PROFESSOR STODOLA'S CONE OF STEAM WEIGHTS



REFERENCE (5)

Figure 5

STODOLA'S ELLIPSE



REFERENCE (6)

$$\frac{W\sqrt{T}}{P} \propto \sqrt{1-\left(\frac{B}{P}\right)^2} \quad \text{, LET K = A CONSTANT}$$

$$\varnothing = \frac{W\sqrt{T}}{P} = K\sqrt{1-\left(\frac{B}{P}\right)^2} = K\sqrt{1-X^2}$$

$$\varnothing^2 = K^2\left(1-X^2\right)$$

$$\frac{\varnothing^2}{K^2} + \frac{X^2}{1} = 1$$

The experimentally determined value of

$$\frac{n + 1}{n} = 2$$

makes relation (D) identical with relation (C), and occurs when n = 1. These relations might be used to improve the accuracy of the proportionality in the future, using the isentropic exponent relation in (E).

Derivation of Codable Equations - The proportionality in (C) can be restated as

$$\frac{\Phi_i}{\Phi_{iD}} = \frac{\sqrt{1 - \left(\frac{B_i}{P_i}\right)^2}}{\sqrt{1 - \left(\frac{B_{iD}}{P_{iD}}\right)^2}} \tag{F}$$

where the subscript "D" refers to the "design set" of the four variables W, P, v and B for form (A) of the flow coefficient or W, P, T and B for form (B). This set is determined from any known load where all flows and pressures in each group are established in relation to each other "by design". In the initial cycle design, this is usually the most efficient operating condition where the pressures of all extractions are set to give nearly equal feedwater temperature rise. For conventional utility cycles this is the "valves wide open" condition, but in cogeneration cycles the "design load" may be as low as fifty percent of maximum.

The "design set" may also be any set determined from the manufacturer's load point heat balances, to permit segmental matching as described earlier (Figure 1).

By algebraic rearrangement and reduction of equation (F) the following results:

$$P_i = \frac{B_i}{\sqrt{1 - \Phi_i^2 \, Y_{iD}}} \tag{G}$$

$$\text{or } B_i = P_i \sqrt{1 - \Phi_i^2 \, Y_{iD}} \tag{H}$$

$$\text{where: } Y_{iD} = \frac{P_{iD}^2 - B_{iD}^2}{P_{iD}^2 \, \Phi_{iD}^2}$$

Y_{iD} is constant for all loads but may be scheduled to match row-by-row calculations as previously described in Figure 1. The flow coefficient Φ_i and Φ_{iD} may be either of the forms in equation (A) or (B) and is the same as that used in programs based on Constant Flow Coefficient, which should facilitate application of these relations.

Equation (G) permits solving for the group inlet pressures "backwards", starting with the known fixed back-pressure B_1, in Figure 3, and working upstream.

For each iteratively established set of flows W_1, W_2 and W_3 for the uncontrolled segment, a new set of pressures P_1, P_2 and P_3 would be calculated for the next iteration.

Equation (G) is, of course, not explicit since Φ_i contains P_i. Further reduction is necessary depending upon the form of Flow Coefficient, (A) or (B). For the "volume" form (A) a quadratic equation

$$P_i^2 - W_i^2 v_i Y_{iD} P_i - B_i^2 = 0 \tag{I}$$

results, so that

$$P_i = \frac{W_i^2 v_i Y_{iD} + \sqrt{W_i^4 v_i^2 Y_{iD}^2 + 4B_i^2}}{2} \tag{J}$$

In using the quadratic formula to derive Equation (J), the sign before the radical must be positive because a negative sign will always give a negative pressure (squared first term).

For the "temperature" form of Flow Coefficient (B) the derivation is much simpler resulting in

$$P_i = \sqrt{W_i^2 T_i Y_{iD} + B_i^2} \tag{K}$$

The temperature form of Flow Coefficient always seems to result in much simpler algebraic formulations and has been recommended as sufficiently accurate for early design work on cogeneration steam cycles, Reference (7), in high back-pressure, superheated steam.

Solving the group inlet pressures "backward" as in Figure 3 and Equations (G), (J) and (K), is usually necessary during some part of the cycle analysis, but is the reverse of the iterative progression in some heat balance codes. The form in equation (H) allows solving "forward" for progressive back-pressures, working downstream in each uncontrolled segment, starting with an assumed value for P_3 in Figure 3. This, of course, requires an iteration or control on the known final back-pressure B_1, in each segment.

It will be noted that for forward progression, equation (H) is <u>explicit</u>, requiring no further reduction. Simple reductions do result for each form of Flow Coefficient, however. For the volume form, (A),

$$B_i = P_i \sqrt{1 - \frac{W_i^2 v_i}{P_i} Y_{iD}} \tag{L}$$

and for the temperature form, (B)

$$B_i = \sqrt{P_i^2 - W_i^2 T_i Y_{iD}} \tag{M}$$

Equation (M) has been successfully used by Syntha Corporation in converging on multi-group segments using the forward progression technique.

<u>Total and Static Pressures</u> - The h-s plot in Figure 3 shows that these relations assume that the leaving velocity from the last stage of each group is completely dissipated, in order that the exit static pressure from each group is the same as

the inlet total pressure to the next group. This, of course, is not quite correct, but the error introduced is expected to be small. In further work using dynamic similarity relations to determine leaving energy, it is planned to refine this approach.

Effect of Number of Stages - The elliptical proportionality is generally applicable for groups consisting of a large number of stages. Mallinson and Lewis, Reference (8), have analysed the deviation due to small number of stages. Their results are presented in Figure 6. Based on these data, Csanady, Reference (4), recommends that for few stages, B_i in equations (G), (J) and (K) should be replaced by

$$B_i - P_{ti}$$

where P_{ti} is the pressure in the throat of an isentropic expansion to sonic from P_i. Choking pressure relations are obtainable from the ASME 1967 Steam Tables, and the modification has been successfully used to determine single stage inlet pressures in a recent design.

Heat Balance Solution Methods - In integrating solution of multi-group segments with heat balance iteration schemes, coding should be internal. Establishment of pressures has a strong effect on convergence stability, and attempts to achieve solutions using PEPSE's operational variables to adjust pressures were unsuccessful. Convergent solutions were achieved internally using SYNTHA III. Examples of complete cogeneration cycles are presented in Figures 7 and 8 which have been analysed with this program.

Graphical Interpretation - A comparative schematic of typical pressure distribution in a three group segment, like Figure 3, is shown in Figure 9. This shows that compared to the linear relations predicted by Constant Flow Coefficient, the curves predicted by the elliptical analogy culminate theoretically at zero flow in the fixed back-pressure B_1.

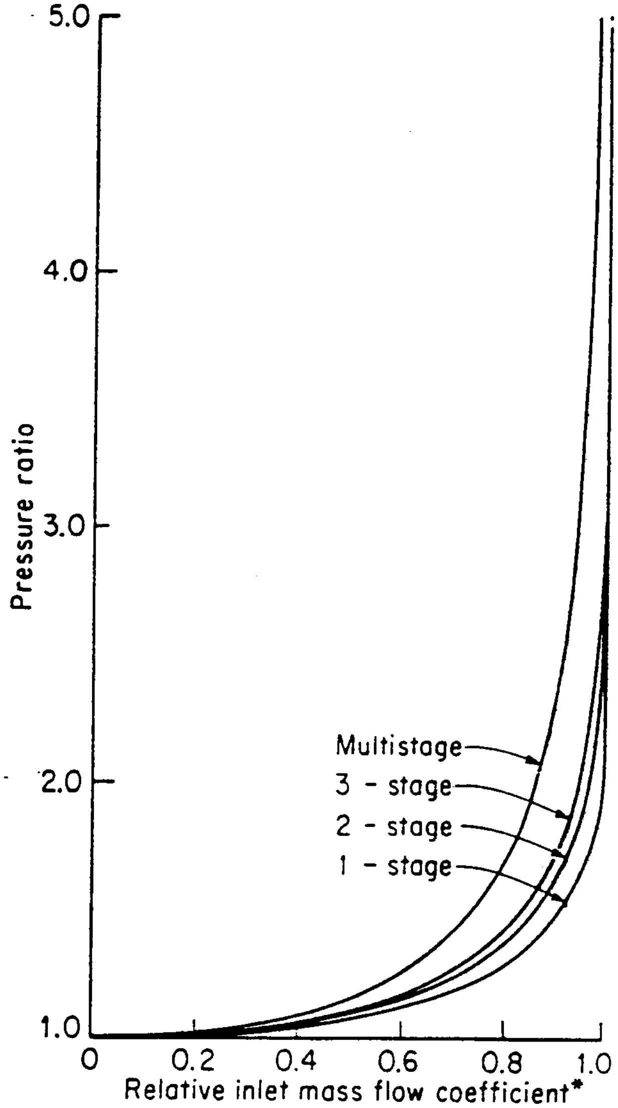
Last Few LP Stages - In the last few LP stages of an uncontrolled expansion to high vacuum the effect of the back pressure becomes very signifiant. If the back pressure is increased, for example, the Constant Flow Coefficient method does not account for the increase in upstream pressures which may actually take place. In a recent analysis using PEPSE, a pressure rise was generated by the program across the last LP group. The situation occurred when abnormally high condenser pressure resulted from by-pass of hot steam during a load trip. The problem is visualized in Figure 10, where it is shown that the Constant Flow Coefficient can predict too low a pressure in stages immediately upstream of the condenser. Whereas the expansion from A to C is positive, a negative expansion, from A to B, resulted at 50 percent load when the backpressure was increased (scheduled as a function of condenser heat load). The pressures were manually corrected using the elliptical relations. This illustrates how the pressure prediction in the last few stages of an uncontrolled expansion could be improved by using Stodola's ellipse. For higher stages, exactly the same pressures were obtained with the elliptical equations as with Constant Flow Coefficient.

SUMMARY AND CLOSURE

Equations and derivations have been given for the nozzle analogy known as the Law of the Ellipse which allows reasonably accurate pressure-flow prediction for high back-pressure and controlled expansions. Work is now in progress at Bechtel on modeling efficiencies using the dynamic similarity method of Reference (9).

Figure 6

EFFECT OF NUMBER OF STAGES



REFERENCE (8)

***FRACTION OF CHOKING FLOW COEFFICIENT**

Figure 7

REFINERY PROCESS UTILITY SYSTEM
PEAK ELECTRICAL LOAD (330 MW) COKE FIRED

JOB 13339-070

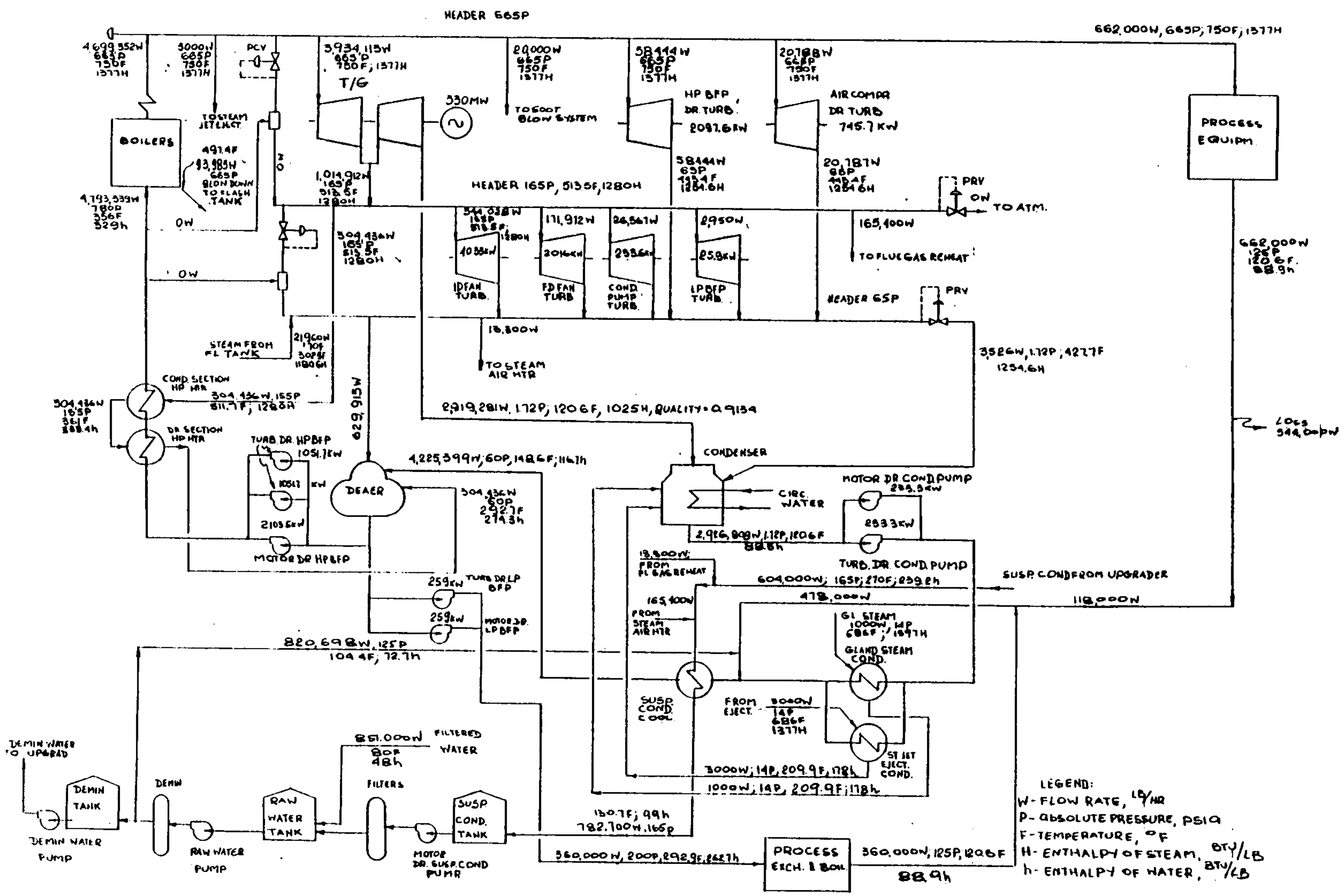


Figure 8

COAL GASIFICATION PROCESS UTILITY SYSTEM
100% LOAD, WINTER (115 MW)

JOB 13650-059

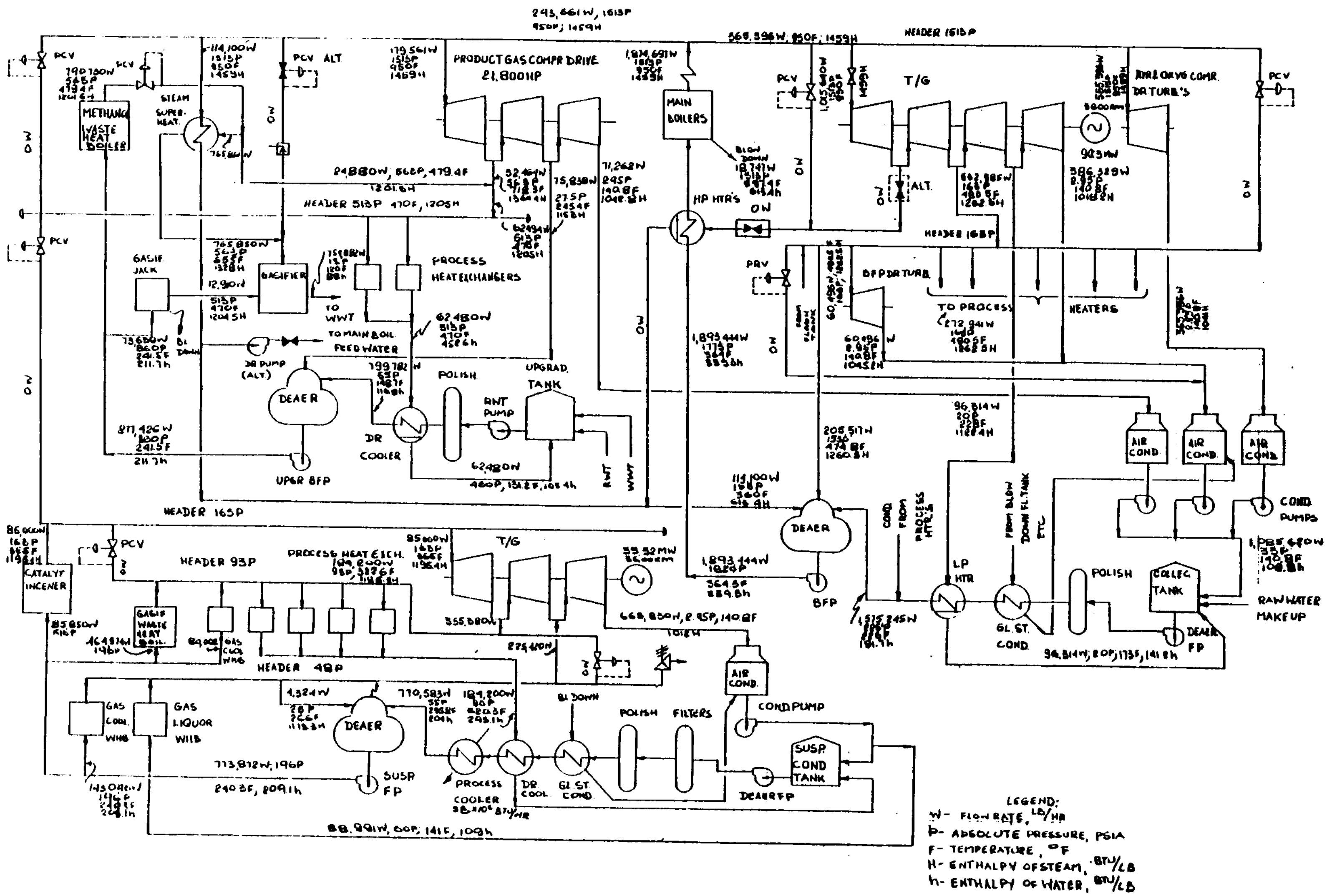


Figure 9

PRESSURE VS. FLOW
FOR HIGH BACK-PRESSURE

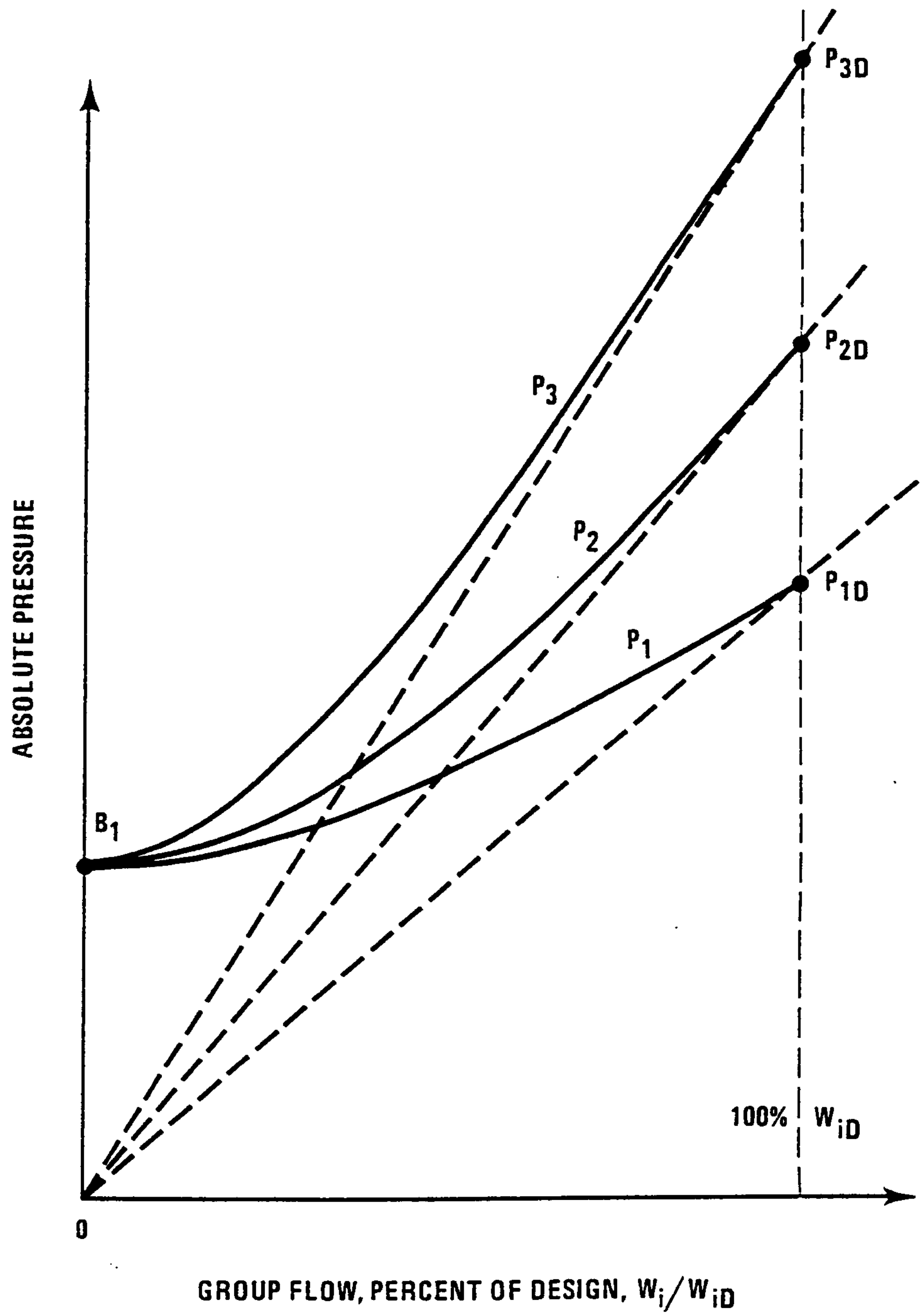
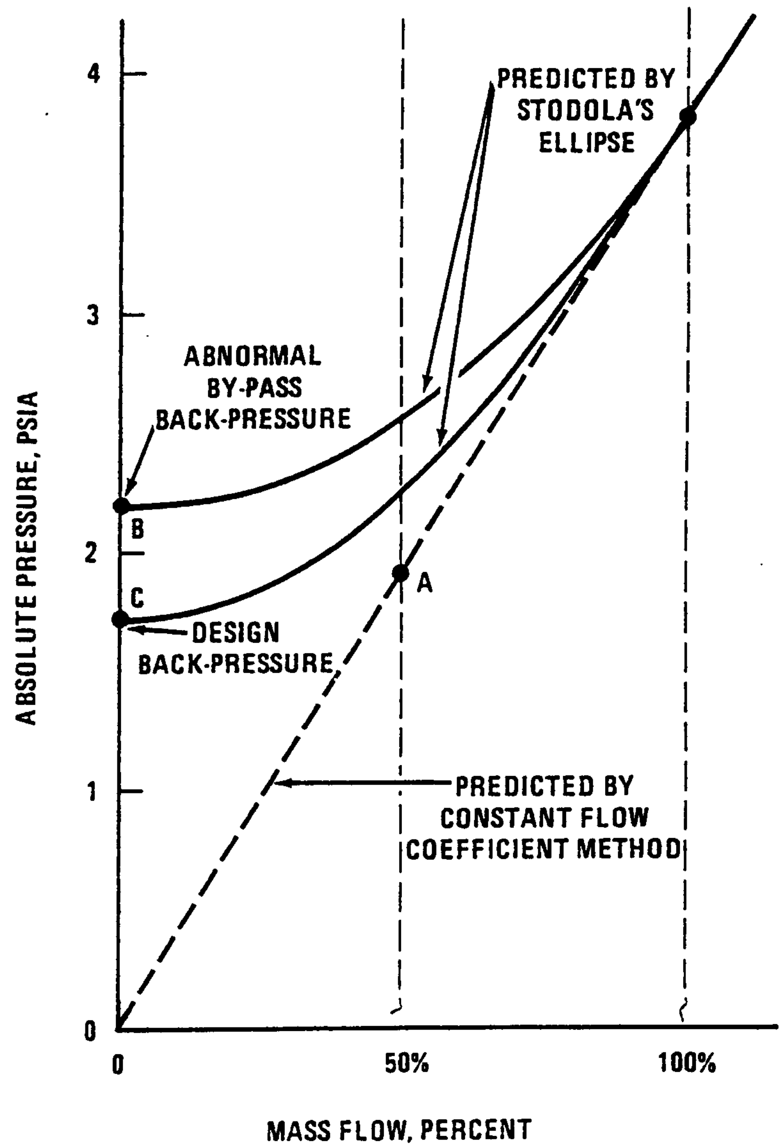


Figure 10

PRESSURE VS. FLOW
AT LAST LP EXTRACTION (MR ZONE)
LARGE NUCLEAR PWR SYSTEM



This method uses the dynamic similarity parameters specific speed and specific diameter to correlate design and off-design efficiencies for optimum turbine designs, or those of any turbine manufacturer. An example of the optimum correlation for single stage designs is shown in Figure 11. The ability to model both pressures and efficiency, independent of any manufacturer's specific technique, but consistent and adaptable to all, should enhance an owner or engineer's capability to design and optimize power plant cycles.

ACKNOWLEDGEMENTS

The assistance of John Bayers and Duane Cao of the Bechtel Mechanical Engineering Staff in development of the relations described in this paper is gratefully acknowledged.

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(2) Ibid, p. 6, para. III, 1.

(3) Energy Incorporated, PEPSE Manual, Vol. 1, User Input Description, Rev. 8, p. 91-B, Equation 5.5-1

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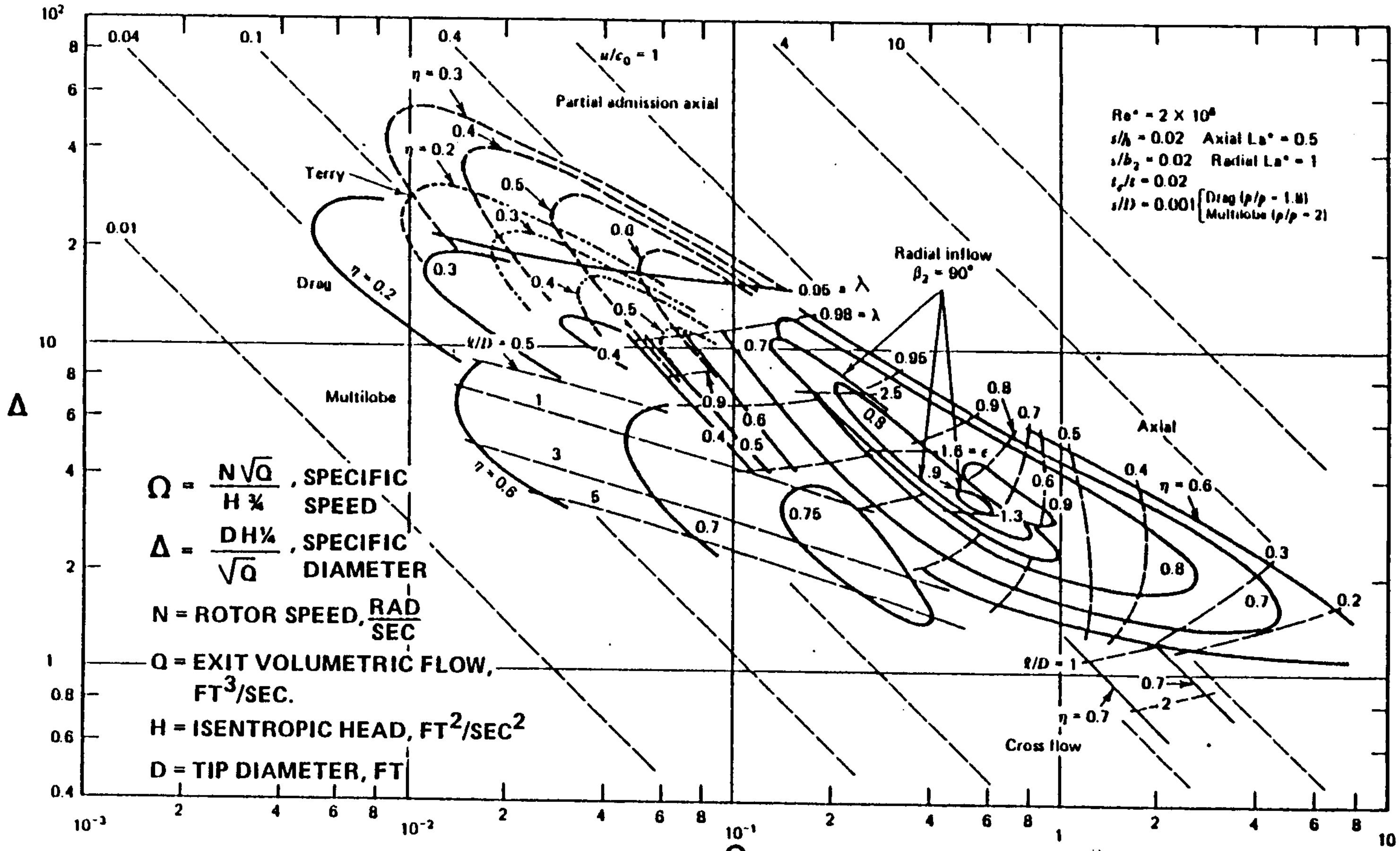
Figure 11

OMEGA–DELTA DIAGRAM FOR
SINGLE STAGE TURBINES AND EXPANDERS



REFERENCE (9)

III. SHORT COURSES

As part of the PEPSE® User's Group Meeting (UGM), four short courses were presented describing various aspects of the PEPSE® code and its use. A survey conducted before the UGM showed that PEPSE® users would benefit from more information about the following: test data evaluation, design mode calculations, heat source modeling, and model debug. As a result, four experts on the PEPSE® code gave short courses for each of these subjects. If you would like to receive a copy of the slides used in the presentation or if you have questions concerning the subject matter of these courses, please direct your inquiries to Roger Kuhl, Energy Incorporated headquarters, (208) 529-1000.

IV. KEYNOTE SPEECH

Unit Improvement Programs at Virginia Electric and Power Company

by

Mr. J. A. Ahladas, VEPCO Manager of
Maintenance and Performance Services

I certainly am pleased to be with you this evening to discuss productivity and some of the programs VEPCO has underway to improve the operation of its generating units....particularly its fossil fuel units.

As some of you may know, I am standing in for Jack Ferguson, our Executive Vice President and Chief Operating Officer. He regrets not being able to be here. A prior commitment made it impossible.

Since Jack Ferguson's arrival at VEPCO, he's been the driving force behind major programs the company has undertaken to increase the productivity of its entire fossil generation system. These programs have been fundamental to a major turnaround in VEPCO's energy supply mix and have resulted in very large fuel savings to the company and our customers.

About a decade ago, just prior to the Arab oil embargo, 50% of the electricity VEPCO generated came from oil-fired units. With the embargo and the subsequent price shocks, this oil-fired capacity turned into an albatross around VEPCO's neck almost overnight. In 1975, we began a major program to convert oil-fired units to coal. To date, we have converted seven units with a combined capacity of 1.8 million kilowatts to coal, making this the nation's largest conversion program. We plan to convert three more units with 436,000 kilowatts of capacity by 1985. This conversion program, combined with a major effort to increase VEPCO's nuclear generation, has resulted in a vastly improved energy supply mix.

Last year, oil accounted for only four percent of our total generation. Nuclear accounted for 41% and coal 37%. If you count the purchases of electricity from some of our neighboring utilities, coal and nuclear power constituted 94% of the company's energy supply in 1982. At the same time, we have undertaken a massive fossil unit improvement program. We are now more than two-thirds of the way through this comprehensive effort to upgrade our entire fossil system, with special attention to our five largest coal units. This improvement program has resulted in significantly improved heat rates and equivalent availability throughout the fossil system and at the five large coal units.

Concurrently, we have completely turned around operations at our Laurel Run Mine, adjacent to VEPCO's Mt. Storm Power Station. In late 1981, we introduced a new longwall miner system at the Laurel Run Mine and we now are producing coal at lower cost than our competitors. These programs have required a major commitment on the part of VEPCO management. But they already are paying rich dividends.

Between 1980 and 1982, VEPCO spent $280 million on the coal conversion and fossil unit improvement programs. During the same period, these programs resulted in $540 million in fuel expense savings for our customers -- nearly two dollars in savings for every one dollar spent. Rather than go into the details of these programs in my remarks, I would like to show you a brief film that covers many of the key elements. This film was prepared last spring for presentation to our stockholders and employees, so it is not technical. But I think it does give a good overall look at what we are doing. Following the film, I will give you some updated information on our results with these programs, and I will be glad to answer any questions you may have. I hope the film gives you some idea of just how comprehensive this effort is. I am happy to say that the improvement cited in the film is continuing.

The equivalent availability for our entire fossil system between the end of 1982 and year to date has increased from 72% to 80%. During the same period, the equivalent availability of our five major coal units went from

64% to 72%. We also are continuing to decrease our heat rates. Between the end of '82 and year to date our overall fossil steam system heat rate dropped an additional one percent. The highest heat rate occurred in 1981: 11,020. In 1982, it was 10,500; and year to date in 1983, it was 10,400. Our fossil steam system fuel expenses have dropped from 2.9 cents per kilowatt-hour in 1980 to 1.9 cents per kilowatt-hour through the first half of 1983.

The most dramatic improvements occurred early in the program due to extensive plant modifications. Now, we are concentrating on operational improvements, such as the use of procedures and computers to monitor individual unit performance and to identify departures from desired conditions. These operational activities will bring smaller but continuing improvements in our system heat rate and unit availability....and to us this means better productivity. The following tables present yearly highlights of VEPCO's unit heat rate improvement program. Also, a group of figures illustrates other major aspects of the program.

UNIT IMPROVEMENT PROGRAM HIGHLIGHTS

1979 - Assigned engineers to root-cause problem identification effort

- Commenced documentation updating and equipment numbering

- Commenced maintenance management program implementation

- Issued Unit Performance Manual

- Commenced staffing of traveling maintenance crews

1980 - Reorganized Production Operations Department by separating nuclear and conventional unit operations

- Conducted extensive outages on large coal-fired units, concentrating on availability-related problems

- Commenced work on numerous capital projects to correct identified problems affecting unit availability

- Implemented cost planning and tracking system

- Continued conversion of units to coal usage and installation of new electrostatic precipitators

- Commenced materials management program

- Commenced administrative controls program for development of uniform procedures

- Commenced more intensive training of station personnel

1981 - Began concentrating on problems affecting heat rate

- Continued capital project work

- Commenced assignment of quality assurance personnel to fossil stations

- Continued previously noted unit improvement work

- Began completing new station facilities (administrative buildings, shops, service buildings, warehouses)

- Began seeing sustained improvement in unit availability

1982 - Completed a number of capital projects (balanced draft conversion of one large coal-burning unit and electrostatic precipitator installations for four other units)

- Implemented preventive maintenance program

- Continued concentration on heat rate improvements and testing

1983 - Expect completion of many projects and programs - essentially complete original scope of unit improvement program

1984 - Cleanup of minor outstanding tasks

- Proceed into a reliability enhancement effort to fine tune unit operations

- Complete converting two additional units to coal