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On the Hybrid Staging of a Lysholm Positive
Displacement Engine with Two Westinghouse Two
Stage Impulse Curtis Turbines

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Donald Alan Parker

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ABSTRACT

The University of California at Berkeley has tested and modeled satisfactorily a hybrid staged Lysholm engine (positive displacement) with a two stage Curtis wheel turbine. The system operates in a stable manner over its operating range (0/1-3/1 water ratio, 120 psia input). Proposals are made for controlling interstage pressure with a partial admission turbine and volume expansion to control mass flow and pressure ratio for the Lysholm engine.

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I	Introduction	1.
	A. Description of Equipment	2.
II	Brief Discussion of Preliminary Data	4.
	A. Discussion of Data Presented	4.
	B. Effect of Interstage Pressure on Power	4.
III	Model and Experimental Data	6.
	A. Model Description	6.
	(1) Lysholm Engine	6.
	(2) Piping and Separator	6.
	(3) Turbine	9.
	B. Model and Experimental Operating Points	11.
IV	Discussion	12.
	A. Justification for a Turbine as Lower Stage	12.
	B. Some Design Considerations	13.
	C. Effect of Well Head Pressure and Quality Degradation and some Proposals for Compensation	13.
	D. Recommendations	16.
V	Summary	17.
VI	Bibliography	18.
VII	Figures	20.
VIII	Calculations	33.

I INTRODUCTION

Steam extracted from geothermal reservoirs has the potential for generating large quantities of motive and/or electric power (Ref. 1). The least common, but more desirable, geothermal reservoirs produce pure steam. In the case of The Geysers, Calif., the steam is put directly into a low pressure turbine to generate electric power. The much more common, but less desirable, geothermal reservoirs produce water or water and steam mixtures. To obtain a suitable turbine steam, the people in Wairakei, New Zealand designed equipment to flash and separate steam from the geothermal reservoir before passing the steam through a low pressure turbine.

It has been proposed that instead of flashing and separating for pure turbine steam a suitable engine be used to accept the "total flow" from these geothermal reservoirs as the engine's working fluid. By accepting the total flow, potentially more usable energy can be obtained from these wet geothermal reservoirs, fig. 1 (Cal. 1). To these ends, a Lysholm positive displacement engine (Ref. 2) has been selected for the application. The engine is singularly rugged and has found its principal application as an air compressor. Two helical screw rotors (male & female) intermesh forming one chamber boundary with the case and rotors forming second and third chamber boundaries, and the end plates forming the final chamber boundary. The end plates also control intake to exhaust volume ratio (5.31/1) fig. 2 (Ref. 3,4,5,6,7,8,9,10).

A. DESCRIPTION OF EQUIPMENT

In order to prove this concept, U. C. Berkeley borrowed just such an engine (a modified Gardner Denver air compressor) from Lawrence Livermore Laboratory and connected it into the campus steam supply (LLNL was also doing such studies). A steam to water heat exchanger was installed to supply heated water for mixture with steam to give a simulated geothermal working fluid ($h_{\text{fluid}} = 1182 \text{ Btu/lb to } 456 \text{ Btu/lb}$). Suitable control and data acquisition equipment were also installed, Fig 3.

Because of the inherent design limitations of the Lysholm engine (Ref. 8), staging is necessary for the differential pressure available on campus (120-5psia). In our case, it was felt that for the large volume flows associated with low pressures, a turbine would be a more efficient lower stage than a Lysholm engine. It should be noted that whether this lower stage should be a turbine or Lysholm has still not been determined based on either an energy consideration or return on investment. In addition, U.C. Berkeley just happens to have two very nice Curtis wheel turbine/generator sets complete with condenser, air ejector and controls, fig. 4 a & b (Ref. 11).

Pure steam is required in the turbines. Therefore, a separator is required between the Lysholm and turbine to remove water. A New Zealand type separator was selected because of its known performance in a geothermal environment, fig. 5 (Ref. 12, 13). Our system was run such that

the pressure in the separator was greater than 14.7psia to avoid pumping the separated water back to atmospheric pressure for disposal. To minimize water reentrainment, the separator was run with a minimum of water in the separator body and, where possible, input steam velocities were kept below 190 ft/sec. The separator fed steam to the turbine at greater than 98% quality for steam flow rates of less than 0.6 lbm/sec and water/steam ratios less than 3/1.

Fig. 6 shows the system in its entirety. Note that the two turbines are run in parallel. Steam and water flow rates are controlled by an operator remote from the Lysholm engine as are the current and voltage for the Eddy brake. Remote turbine controls (field current, turbine load, and condenser pressure) and the data acquisition system are also located in the same control booth.

It is desired to predict the system operating parameters and their relationship between one another. As a preliminary to a description of the system analytical model, some preliminary test results relating interstage pressure to turbine steam flow will be given. It is the specific intent of this discussion to introduce some of the component relationships and their effect on one another, so that the analytical model will make more sense. The model will then be used to predict actual system performance. A general discussion will follow.

II BRIEF DISCUSSION OF PRELIMINARY DATA

A. DISCUSSION OF DATA PRESENTED

A series of tests were done to determine the performance of the separator. The preliminary test results are shown in fig. 7. The Lysholm to separator curves describe the flow related pressure drop of the two phase or single phase fluid from the Lysholm engine to the separator at various water/steam ratios vs. the mass flow rate of steam to the turbine. Also shown is the separator pressure minus 14.7psia required for a particular steam mass flow rate through the turbine. So, if we wish to know the Lysholm exhaust pressure for a separator water/steam ratio of 1/1 and turbine steam mass flow of 0.5 lb/sec we have

$$\begin{aligned} P_{\text{Lysholm exhaust}} &= 14.7\text{psia} + P_{\text{turbine}} + P_{\text{sep @ 0.5 lb/sec}} \\ &\qquad\qquad\qquad 1/1 \text{ w/s} \\ &= 14.7\text{psia} + 4.5 \text{ psi} + 6 \text{ psi} \\ &= 25.2\text{psia} \end{aligned}$$

B. EFFECT OF INTERSTAGE PRESSURE ON POWER

Generally, the pressures between the Lysholm engine and the turbine are referred to as the interstage pressure. Varying this pressure changes the power realized from the engine and the turbine.

If we examine a turbine curve, (Willans line), an increase in mass flow rate increases the power output from the turbine, but because the steam flow rate through

the turbine is choked, the interstage pressure also increases, fig. 7. The Lysholm engine follows a similar power vs flow rate curve (Willans line) to the turbine. But, for a fixed Lysholm input quality, an increase in mass flow rate corresponds to an increase in Lysholm engine RPM. For an increase in interstage pressure caused by higher parasitic losses in the interstage piping and turbine pressure increase due to higher steam flow, the power increase in the Lysholm engine due to the increase in mass flow (RPM) can be largely offset by the reduced pressure ratio across the Lysholm. Depending on the operating point, both increasing or decreasing mass flow rate can decrease system power output, depending upon the turbine and Lysholm performance characteristics.

Continuing the analysis shows that the greatest power occurs at maximum steam flow to the turbine and the minimum interstage pressure. By examining the Lysholm 9000 RPM line (Lysholm drive belt maximum RPM), the minimum interstage pressure always occurs for pure steam. This point, pure steam and 9000 RPM Lysholm speed, represents the maximum power output from this system.

III MODEL AND EXPERIMENTAL DATA

A. MODEL DESCRIPTION

The analytical model is composed of three parts; (1) Lysholm engine, (2) piping and separator, and (3) turbine. Each part will be briefly discussed here. A complete calculation (Cal 2) shown in the appendix gives the formulas for each part. Please refer to it for a detailed methodology of the model.

(1) LYSHOLM ENGINE

The Lysholm engine model was done by Karl Brown (Ref. 8). The engine model was developed by curve fitting a data matrix of engine operating conditions (inlet quality, pressure ratio and engine RPM) to those desired outputs (power, exhaust quality, mass flow rate, and efficiency). Knowing the three operating conditions uniquely defines the desired output. Because the piping and turbine models affect the pressure ratio, this unfortunately precludes specifying an operating condition and necessitates trial and error solution.

(2) PIPING AND SEPARATOR

The piping and separator model can be broken into two sections; (1) from Lysholm to separator and (2) from separator to turbine.

Taking the second section first, we wish to calculate the pressure drop for a given steam mass flow

rate through a series of pipes and fittings. Calculating the Mach number (M) (Cal 3), in the worst case, yields $M=0.15$. Schlichting (Ref 15) says that a gas can be treated as incompressible if $M < 0.32$. The Mach number falls below this limit, so we can proceed by standard pipe pressure drop calculations for incompressible flow. To do this, we first find the Reynolds number using the mass flow rate and saturation steam properties evaluated at the turbine chest pressure from the turbine model. The pipe roughness factor (ϵ/D_e) is found and with the Reynolds number we proceed to the Moody chart for a value of friction factor (f) (Ref. 16). The values of (L/D & K) (Ref. 16) for the various fittings and pipe are summed and with the proceeding values of f and velocity (V) are substituted into the pressure difference.

$$\Delta P = \left(f \frac{L}{D} + K \right) \frac{V^2}{2g}$$

This pressure difference plus the turbine chest pressure yields the separator pressure. In its operational range, (0/1 - 3/1 water/steam, $\dot{m}_s < 0.6$ lb/sec), the New Zealand separator separates a water/steam mixture, (accepting water/steam from the Lysholm and passing saturated steam to the turbine plus discharging the fluid to the drain).

Two phase flow losses are generally approximated by empirical relations. A modified method developed

by W. R. Tek (Ref) is used to calculate the models first section pressure drop, (this section of pipe carries two phase flow which includes the separator). The pressure drop parameter is found (ratio of single phase pressure drops in two phase flow ($\Delta P_{\text{liquid}} / \Delta P_{\text{gas}})^{1/2}$). This parameter is used to find the correlation factor ϕ for liquid and gas. These factors are squared, then multiplied by their respective single phase pressure drops to obtain the two phase flow pressure drop (actual). Empirically, it has been found that the steam flow pressure drop times its respective ϕ_g steam gives the better of the two two phase flow pressure drops. The single phase pressure drops are calculated as outlined above.

The sum of the two section pressure drops plus the turbine steam chest pressure equals the Lysholm exhaust pressure. The pressure ratio for the Lysholm engine can be determined ($120/P_{\text{exhaust}}$).

The Lysholm exhaust quality is found by starting at the separator. The separator pressure and quality are known, therefore the enthalpy is known. Neglecting heat loss in the piping (Cal. 4) and elevation change, (Cal. 5) we assume a constant isenthalpic compression to the Lysholm exhaust and find the exhaust quality. With pressure ratio and exhaust quality, we can next calculate the Lysholm operating conditions (x_{in} , RPM given $\dot{M}$, x_2 , PR). At this point, we have uniquely

solved for the Lysholm outputs (power, exhaust quality, mass flow rate, and efficiency) and for the interstage pressure drops.

(3) TURBINE

The turbine part of the model is in two parts; the first being the turbine chest pressure and the second the power output. An empirical relationship (Grashof's relationship) is used to find the turbine chest pressure for saturated steam, given the mass flow rate and nozzle area. This chest pressure is used in the piping and separator model for fluid physical properties and in the Lysholm model for pressure ratio and exit quality.

The power output from the turbine is found by using the corrected nozzle velocity and construction of velocity triangles. It is best to refer to the calculation (Cal. 2), as there are diagrams and numbers. Here I will suffice to describe what is going on.

The first step is to calculate the isentropic nozzle velocity, given the chest pressure by Grashof's formula and an exhaust pressure of 5 psi. Chest enthalpy is defined by saturated steam at chest pressure. From this point, an isentropic pressure drop determines the enthalpy at 5 psia. The enthalpy difference between chest pressure and 5 psia is substituted into the isentropic velocity equation. This velocity is multiplied by a coefficient (nozzle losses, ref. 18) to account for non-ideal nozzle characteristics.

The mass flow rate impinges on the turbine blades at this velocity with some incident angle (defined by physical geometry).

The power output from the turbine is defined as the force ($\dot{M} \Delta \vec{V}$) applied to the turbine blade in the direction of rotation times the blade velocity ($\vec{V}_B$). The velocity change through the blades ($\Delta \vec{V}$) is our next task. The corrected nozzle velocity is the starting point. The relative velocity of the first stage is then calculated from the nozzle velocity, incident angle, and blade velocity. This entrance relative velocity is corrected for supersonic flow loss and turning loss to give the exit relative velocity. With the blade exit angle, the vector difference between the entrance and exit velocity in the bucket velocities direction gives the first stage $\Delta \vec{V}$, and when multiplied by the mass flow rate gives the force. The same is done for the second stage. The sum of these forces times the bucket velocity gives the total power output. Disk and bucket losses plus bearing losses are then subtracted, giving the total mechanical power available for electricity generation. This is a bit sketchy. But the intent is to go through calculation #2. This is the extent of the model description. Table 1 shows predicted vs measured power outputs vs quality. Calculation 6 shows the data reduction calculations for the experimental data.

B. Model and Experimental Operating Points

Table 1 shows the system performance, predicted vs observed. The first pass effort to model the system performance predicted to high a turbine power. Soderberg's correlation was substituted for blade loss and nozzle loss coefficients (ref 22). A fudge factor of 5 also had to be multiplied by this correlation for the turbine power to match the observed performance. The second pass effort reflects the changes in the turbine model. The error is estimated to be less than 10%.

The piping model was based on figure 7 data and M.R. Tek's method for pressure drop in two phase flow. The uncertainty in the experimental data (1 psi) shows a 21% difference between measured and predicted values in the high quality range (greater than .5). This error is reduced as the input quality goes down.

IV DISCUSSION

At this point, a discussion is in order about the Lysholm/turbine staged system. It will be followed by a discussion of quality and pressure degradation from the geothermal fluid to the Lysholm engine and schemes for compensating for these changes.

A. JUSTIFICATION FOR A TURBINE AS LOWER STAGE

As stated earlier, a turbine was chosen as the more efficient machine for the lower stage in this system because of the large volume flow rates at sub atmospheric pressures. Large volume flow rates require large flow areas. A turbine is a much more efficient user of frontal area than a Lysholm engine. To compensate for this poor efficiency, the Lysholm's engine diameter is increased. Since one current design limitation is hoop stress, increasing the diameter only aggravates this problem ($\omega^2 r$). Larger diameters also have higher flow losses.

This argument can be further expanded if we consider using two phase flow to subatmospheric expansion in a Lysholm engine or separating and expanding through a turbine. Fig 7 (Cal. 7) shows the break-even Lysholm efficiency for sub-atmospheric expansion vs. input quality. The efficiencies required to match the turbine performance have not been achieved in practice. This is not to say that these efficiencies wouldn't be achieved in practice, but that it is unlikely that any considerable improvement over the conventional turbine will warrant the Lysholm's use as a subatmospheric expander. It should be further noted that the addition of dissolved

gases to this analysis will only tend to increase the breakeven efficiency necessary for the Lysholm, making the turbine more attractive.

B. SOME DESIGN CONSIDERATIONS

The other major point that should be reemphasized is that of parasitic losses between the Lysholm and turbine. The pressure increase associated with increasing mass flow through the turbine develops power. To a "controlled" degree this is beneficial. The pressure increase (flow loss) associated with decreasing quality and increasing mass flow rate in the piping costs power. This is to be avoided at all costs. This is principally a design problem (for example, the machinery should be located close together, as few pipe bends, contraction or expansions, valves, and fittings as possible, low steam velocities (less than 180 ft/sec), minimal length of pipe with two phase flow, proper sizing of equipment etc.) and once recognized can be treated as such.

C. EFFECT OF WELL HEAD PRESSURE AND QUALITY DEGRADATION AND SOME PROPOSALS FOR COMPENSATION.

In the normal life of a geothermal reservoir both quality and well head pressure decrease with use. This presents design problems to the design engineer, for he is being asked to design around a region instead of a point. One option is to specify to the steam supplier an acceptable steam source and let him handle the problem. This option stated, we can now move on to the options available to the design engineer of the equipment.

Although coupled, let us first examine the effect of decreasing quality on the performance of a staged Lysholm/turbine system. Decreasing fluid quality to the Lysholm engine (maintained at a constant speed, generator) increases the total mass flow through the engine. But, if we examine the steam mass flow to the turbine, we find that decreasing quality up to approximately 35% results in less steam to the turbine. Further quality decrease below 35% results in increasing steam to the turbine, fig 8 (Cal. 8).

To some degree, the increased (decreased) mass flow can be compensated for by a partial admission turbine. In our case, not only is the partial admission turbine being used to improve the turbines off design peak performance but also to regulate the interstage pressure to insure optimum Lysholm performance (pressure ratio). Physically, what the partial admission turbine does is to change the turbine's input flow area into the impulse stage. By examining Gra-shof's relationship again

$$\dot{m} = 0.0165 A_o (P_{CHERT})^{0.95}$$

we can see this relationship. There is some efficiency lost in the Curtis wheel impulse stages due to windage and turbulence in the blades as they pass idle nozzles, and under expansion for low mass flow rates. Efficiency is also lost in the reaction stages due to incorrect angles of incidence. Taken as a whole, the overall turbine efficiency is better than for full admission with a reduced

turbine chest pressure (Ref 20).

For maximum output power, the first stage (impulse) can be bypassed and steam fed directly into a reaction stage. Since both the pressure to this stage and area compared to the impulse stage are larger, greater mass flow rates are passed. Even when accounting for high windage loss in the impulse stage, and loss associated with incorrect incident angles, and leaving loss associated with the reaction stage, greater power is still generated.

Decreasing pressure at constant quality to the Lysholm decreases the mass flow through the engine (decreases Lysholm power at same P.R. = 5.31 and constant RPM). Not only does the total mass flow decrease through the engine but also the steam mass flow to the turbine. The turbine compensates for low mass flow rate by increasing steam chest pressure (partial admission reduces input area) which increases the interstage pressure. The Lysholm's performance now is not only lowered by input pressure degradation but also by higher back pressures (pressure ratio going down). It is proposed to compensate for the lower mass flow rate and lower pressure ratio by changing the input to output volume expansion ratio of the Lysholm engine. If, for example, the input port were a slider concentrically or perhaps radially and tangentially mounted, increased the input volume, greater flow (fixed Lysholm RPM) through the engine would result, as well as more steam to the turbine (Cal 9). The turbine would respond by opening nozzles and the Lysholm's pressure ratio would be more nearly in a

range for its new input to exhaust volume ratio. This is, of course, speculation since there is no data available on such a port system. The trends, though, are correct.

D. RECOMMENDATIONS

If further research is to be done to apply such a hybrid system to geothermal use, the following areas should be considered.

(1) The most important task is to gain field experience with the present engine. Scale formation is an important problem not yet considered, as are hardware material compatibility problems with the geothermal fluid. This step cannot be bypassed in lieu of laboratory work.

(2) Further work should be done on characterizing the Lysholm engine at various expansion ratios. A phenomenological model then should be constructed.

(3) Based on the above a variable volume expansion Lysholm and partial admission turbine should be built and tested.

V SUMMARY

University of California at Berkeley has tested and modeled a hybrid staged Lysholm engine (positive displacement) with a two stage Curtis wheel turbine. It was found that the system operates in a stable manner over its operating range (o/l - 3/1 water ratios, 120 psia input). The model satisfactorily predicts the system components, but further work should be done to allow a more flexible two phase pipe flow pressure drop model. Work should also begin on a phenomenological model for the Lysholm engine. Proposals are made for controlling interstage pressure with a partial admission turbine and volume expansion to control mass flow and pressure ratio for the Lysholm engine.

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LYSHOLM BREAK EVEN EFFICIENCY WITH TURBINES VS INLET QUALITY

FIGURE 1
20.

GEOTHERMAL EXPANDER FOR
STEAM-WATER MIXTURE

FIGURE 2 CROSS SECTION OF LYSHOLM ENGINE

REV.	DESCRIPTION	DATE	BY	CHKD.	APP'D.	REVISION
1	AS SHOWN					
2	REVISION					
3	REVISION					
4	REVISION					
5	REVISION					
6	REVISION					
7	REVISION					
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WESTINGHOUSE TURBINE GENERATOR SYSTEM

FIGURE 5

HYBRID GEOTHERMAL POWER SYSTEM

FIGURE 6

FIGURE 7
29.

SYSTEM PERFORMANCE
PREDICTED vs OBSERVED

9000 RPM

LYSHOLM (KW)	TURBINE (KW)	PRESSURE RATIO	INLET QUALITY
25.0	42.0	4.6	1.0
24.7	22.9	4.5	
24.1 (23.1) 23.8	39.6 (23.4) 21.7	4.4 (4.3) 4.4	0.5
22.7	39.8	4.2	0.33
22.4	21.8	4.2	
20.8	41.1	4.0	0.25
20.5	22.4	4.0	

TABLE 1

30.

xx.x First pass through
(xx.x) Experimental data
xx.x Second pass through

LYSHOLM SUBATMOSPHERIC EXPANSION
BREAK-EVEN EFFICIENCY vs QUALITY

FIGURE 8

STEAM MASS FLOW RATE vs
INLET QUALITY (9000 RPM)

FIGURE 9

CALCULATION #1

MECHANICAL EFFICIENCY OF LYSIOLM AND TURBINE
ENERGY BREAKDOWN VS QUALITY AND ENTHALPY

P_1 = WELL HEAD PRESSURE
 P_2 = SEPARATOR OUTPUT PRESSURE
 P_3 = LYSIOLM/TURBINE EXHAUST PRESSURE

— LYSIOLM EXPANSION $P_1 - P_3$
— TURBINE EXPANSION $P_2 - P_3$

$P_2 = 90 \text{ psi}$ $h_{2T} = \text{SATURATE}$
 $P_3 = 15 \text{ psi}$ $x' = \text{QUALITY ON EXPANSION FROM } T_1 \text{ TO } P_2$

(ENERGY BALANCE - LYSIOLM = TURBINE)

$$(1) \quad \eta_L \left[(x h_{fg} + h_g) - h_{3L} \right] m_T$$

$$= \eta_T \left[(h_{2T} - h_{3T}) \right] m_{\text{TOT}} x'$$

(TURBINE EXHAUST ENTHALPY)

$$h_{3T} = h_g + x h_{fg}$$

$$x = \frac{s_2 - s_{3T}}{s_{fgT}} = \frac{1.6113 - 0.3137}{1.4415} = 0.9002$$

$$(2) \quad \text{(TURBINE EFFICIENCY)} \quad \eta_T = 70\%$$

$$= (181.21 + 969.1 \times 0.9002) = 1054.1 \text{ Btu/lb}$$

(ENERGY BALANCE)

$$(2) \quad \eta_L \left[(x h_{fg} + h_g) - h_{3L} \right] = 0.7 \left[1185.3 - 1054.1 \right] x'$$

(LYSIOLM BREAKDOWN EFFICIENCY) $= 91.8 x'$

$$(3) \quad \eta_L = \left[\frac{91.8 x'}{\left[(x h_{fg} + h_g) - h_{3L} \right]} \right]$$

(SEPARATOR QUALITY)

$$x' = \left[\frac{h_1 - h_{f90psi}}{h_{f90psi}} \right] = \left[\frac{h_1 - 290.7}{894.6} \right]$$

(QUALITY AT 15 psi)

$$x'' = \left[\frac{s'_1 - s'_{f15psi}}{s'_{f15psi}} \right] = \left[\frac{s'_1 - 0.3137}{1.4415} \right]$$

(LYSHOLM EXHAUST ENTHALPY)

$$h_{2L} = [h_{f3} + x' h_{f3}] = [181.21 + x' 969.7]$$

x_1	h_1 Btu/lb	P_1 psia	x'	x''	h_{2L} Btu/lb	m_L	m_{LMT}
0.05	421	260	0.15	0.21	387.1	0.41	
0.1	451.5	230	0.18	0.24	411.7	0.42	(41.6 - 0.17)
0.3	608.3	200	0.36	0.37	538.1	0.47	(42.2 - 0.15)
0.5	768.6	170	0.53	0.51	671.8	0.50	(41.3)
0.7	932.6	140	0.72	0.65	814.3	0.56	(41.8)
0.9	1100.6	110	0.91	0.81	968.3	0.63	
1	1185.3	90	1	0.90	1094.1	0.70	(36.7 - 1)

LYSHOLM WENT M (%)

APRIL 26, 1982

CALCULATION # 2

TRIAL CALCULATION FOR SYSTEM MODEL

TURBINE

$$\dot{m}_{\text{TURBINE}} = 0.55 \text{ lb/sec (STEAM MASS FLOW - GIVEN)}$$

$$P_{\text{CHEST}} = \left[\dot{m}_T / 0.0165 A_c \right]^{1/0.97} \quad (\text{GRASHOF'S RELATIONSHIP FOR TURBINE CHEST PRESSURE})$$

$$= \left[0.55 \text{ lb/sec} / (0.0165 \times 1.766 \text{ in}^2) \right]^{1/0.97}$$

$$= 20.67 \text{ PSIA}$$

$$h_{\text{CHEST}} = 1156.8 \text{ Btu/lb} \quad s_{\text{CHEST}} = 1.7298 \text{ @ } 20.67 \text{ PSIA} \quad (\text{CHEST ENTHALPY ENTROPY})$$

$$\text{@ } 5 \text{ PSIA} \quad s_{\text{CHEST}} \quad x = \frac{1.7298 - 0.2349}{1.6094} = 0.929 \quad (\text{EXHAUST QUALITY})$$

$$h_{\text{EXHAUST}} = 1060.0 = (130.20 + 0.929 \times 1000.9) \text{ Btu/lb} \quad (\text{EXHAUST ENTHALPY})$$

$$\Delta h_s = (1156.8 - 1060.0) = 96.8 \text{ Btu/lbm} \quad (\text{ISENTROPIC } \Delta h)$$

$$V_s = \sqrt{2 J g \Delta h_s} \quad (\text{ISENTROPIC VELOCITY})$$

$$= \sqrt{2 \times 778.26 \frac{\text{ft} \cdot \text{lb}_f}{\text{lb}_m} \times 32.2 \frac{\text{ft}}{\text{sec}^2} \times 96.8 \text{ Btu/lbm}}$$

$$= 2203 \text{ ft/sec}$$

$$V_{\text{NOZZLE}} = V_s \times \left[0.96 - \left(\frac{V_s}{4430} \right)^{7.1} \right] \quad (\text{ACTUAL NOZZLE VELOCITY})$$

$$= 2203 \times \left[0.96 - \left(\frac{2203}{4430} \right)^{7.1} \right] \text{ ft/sec}$$

$$= 2099 \text{ ft/sec}$$

$$V_{\text{BLADE}} = \text{RPM} \times 2\pi r \quad (\text{BLADE VELOCITY})$$

$$= 3600 \text{ RPM} \times 2 \times \pi \times \frac{10.186 \text{ in}}{12} \times \frac{1 \text{ min}}{60 \text{ sec}}$$

$$= 320 \text{ ft/sec}$$

V_{1R} = (FIRST STAGE RELATIVE ENTRANCE VELOCITY)

$$V_{1R} = \left[(V_N \cos \alpha - V_B)^2 + (V_N \sin \alpha)^2 \right]^{\frac{1}{2}}$$

$$= \left[(2099 \cos 15 - 320)^2 + (2099 \sin 15)^2 \right]^{\frac{1}{2}}$$

$$= 1792 \text{ ft/sec}$$

V_{2R} = (FIRST STAGE RELATIVE EXIT VELOCITY)

$$= V_{1R} \times 0.91 \left[1 - \left(\frac{V_{1R}}{4070} \right)^{0.42} \right]$$

$$= 1792 \times 0.91 \left[1 - \left(\frac{1792}{4070} \right)^{0.42} \right]$$

$$= 1629 \text{ ft/sec}$$

V_2 = (FIRST STAGE EXIT VELOCITY)

$$= \left[(V_{2R} \cos \beta - V_B)^2 + (V_{2R} \sin \beta)^2 \right]^{\frac{1}{2}}$$

$$= \left[(1629 \cos 19 - 320)^2 + (1629 \sin 19)^2 \right]^{\frac{1}{2}}$$

$$= 1331 \text{ ft/sec}$$

V_3 = (EXIT VELOCITY OF STATOR)

$$= V_2 \times 0.92 \left[1 - \left(\frac{V_2}{4070} \right)^{0.42} \right]$$

$$= 1331 \times 0.92 \left[1 - \left(\frac{1331}{4070} \right)^{0.42} \right]$$

$$= 1224 \text{ ft/sec}$$

V_{3R} = (SECOND STAGE RELATIVE ENTRANCE VELOCITY)

$$= \left[(V_3 \cos \gamma - V_B)^2 + (V_3 \sin \gamma)^2 \right]^{\frac{1}{2}}$$

$$= \left[(1224 \cos 20 - 320)^2 + (1224 \sin 20)^2 \right]^{\frac{1}{2}}$$

$$= 974 \text{ ft/sec}$$

$$\begin{aligned}
 V_{4R} &= (\text{SECOND STAGE RELATIVE EXIT VELOCITY}) \\
 &= V_{3R} \times 0.93 \left[1 - \left(\frac{V_{3R}}{4070} \right)^{8.42} \right] \\
 &= 974 \times 0.93 \left[1 - \left(\frac{974}{4070} \right)^{8.42} \right] \\
 &= 865 \text{ ft/sec}
 \end{aligned}$$

$$\begin{aligned}
 V_4 &= (\text{SECOND STAGE EXIT VELOCITY}) \\
 &= \left[(V_{4R} \cos 20 - V_3)^2 + (V_{4R} \sin 20)^2 \right]^{1/2} \\
 &= \left[(865 \cos 20 - 320)^2 + (865 \sin 20)^2 \right]^{1/2} \\
 &= 575 \text{ ft/sec}
 \end{aligned}$$

$$\begin{aligned}
 C_{ws} &= (\text{WET STREAM LOSS}) \\
 &= \left(\frac{P_{in}}{P_{out}} \right)^{0.042} \\
 &= \left(\frac{120}{20.67} \right)^{-0.042} \\
 &= 0.93
 \end{aligned}$$

$$\begin{aligned}
 F &= (\text{FORCE APPLIED TO ROTOR}) \\
 &= \dot{m} \left[(V_1 \cos \alpha_1 - V_R) + (V_{2R} \cos \beta) \right. \\
 &\quad \left. + (V_3 \cos \gamma - V_R) + (V_{4R} \cos \delta) \right] \\
 &= 0.55 \left[(2099 \cos 15 - 320) + (1629 \cos 19) \right. \\
 &\quad \left. + (1224 \cos 20 - 320) + (865 \cos 20) \right] \\
 &= 2690 \frac{\text{ft} \cdot \text{lb}}{\text{sec}^2} \times \frac{1}{32.2} \frac{\text{lb}_f}{\text{lb}_m} \frac{\text{sec}^2}{\text{ft}} \\
 &= 84 \text{ lb}
 \end{aligned}$$

$$\pi_T' = (\text{TURBINE POWER TOTAL})$$

$$= F \times V$$

$$= 8416 \times 3600 \text{ RPM} \times 2 \text{ ft} \times \frac{10.186}{12} \frac{1 \text{ Kw}}{44,240 \frac{11 \text{ lb}}{1 \text{ min}}}$$

$$36.5 \text{ Kw}$$

$$\pi_T = \pi_T' \times C_{ws} = (\text{TURBINE POWER ACTUAL})$$

$$= 36.5 \text{ Kw} \times 0.93$$

$$= 33.9 \text{ Kw}$$

$$\pi_L = (\text{TURBINE POWER LOSSES})$$

$$= 0.500 \text{ Kw} \quad (\text{BEARINGS})$$

$$= 0.042 D^2 p \left(\frac{u}{100} \right)^{2.9} \times 2 \quad (\text{DISC DRAG})$$

$$= 0.042 \times \left(\frac{20.4}{12} \right)^2 \times \left(\frac{1}{73.53} \frac{\text{lb}}{\text{ft}^3} \right) \times \left(\frac{320}{100} \right)^{2.9} \times 2$$

$$= 0.096 \text{ Kw}$$

$$= 0.187 D p \ell^{1.25} \left(\frac{u}{100} \right)^{2.9} \times 2 \quad (\text{BUSHET LOSS})$$

$$= \left[\left(0.187 \times \left(\frac{20.4}{12} \right) \times \left(\frac{1}{73.53} \right) \times \left(\frac{320}{100} \right)^{2.9} \right) \times \left((0.46)^{1.25} + (0.86)^{1.25} \right) \right] \times 2$$

$$= 0.304 \text{ Kw}$$

$$\sum \pi_L = 0.9 \text{ Kw} \quad (\text{SUM OF TURBINE LOSSES})$$

$$\pi_{\text{actual}} = (33.9 - 0.9) = 33 \text{ Kw}$$

(Actual Turbine Power)

NOTE: ELECTRICAL LOSSES

$$\begin{aligned}
 L_{wf} &= 1.023 \times 10^{-4} (\text{RPM})^{1.273} \times 2 \quad (\text{WINDAGE \& DRAG}) \\
 &= 1.023 \times 10^{-4} (3600)^{1.273} \times 2 \\
 &= 6.9 \text{ kW}
 \end{aligned}$$

$$\begin{aligned}
 L_{cr} &= 0.1517 I^{2.346} \quad (\text{CORE \& RESISTANCE}) \\
 &= 0.1517 ((1)^{2.346} + (2)^{2.346}) \\
 &= 1.0 \text{ kW}
 \end{aligned}$$

PIPE \& SEPARATOR

TURBINE TO SEPARATOR

(GIVEN DATA FROM TURBINE MODEL)

$$p_{\text{chest}} = 20.07 \text{ psia} \quad T = 229.5^\circ \text{F SAT}, \quad v = 19.66 \frac{\text{ft}^3}{\text{lb}}$$

$$\begin{aligned}
 RE &= \frac{VD}{\nu} = \left[\frac{0.55 \times 19.66 \frac{\text{ft}}{\text{SEC}}}{1\pi (4/12 \times 12)^2} \right] \left[\frac{4/12 \text{ ft}}{0.82 \times 10^{-5} \times 19.66 \frac{\text{ft}^2}{\text{SEC}}} \right] \\
 &= 2.56 \times 10^5 \quad (\text{REYNOLDS NUMBER})
 \end{aligned}$$

$$\frac{\epsilon}{D} = 0.00044 \Rightarrow f = 0.018 \quad (\text{ROUGHNESS COEFF})$$

$$\begin{aligned}
 \Delta p_f &= f \frac{L}{D} \frac{V^2}{2g} \quad (\text{PIPE } \Delta P) \\
 &= 0.018 \times \left(\frac{50}{4/12} \right) \times \frac{(124.04 \text{ ft/sec})^2}{2 \times 32.2} \times \frac{1}{144 \times 19.66} \\
 &= 0.23 \text{ psi}
 \end{aligned}$$

$$\begin{aligned}
 \Delta p_{\text{minor}} &= f \frac{L}{D} \frac{V^2}{2g} \quad (\text{MINOR PIPE } \Delta P) \\
 &= 0.018 \times (231) \times \frac{(124 \text{ ft/sec})^2}{2 \times 32.2} \times \frac{1}{144 \times 19.66}
 \end{aligned}$$

$$= 0.35 \text{ psi}$$

$$\begin{aligned}\Delta p &= \Delta p_{\text{pipe}} + \Delta p_{\text{minor}} \quad (\text{TOTAL } \Delta p) \\ &= 0.23 + 0.35 \\ &= 0.58 \text{ psi}\end{aligned}$$

SEPARATOR TO LYSITOLM (M.R.TEK method)

$$\begin{aligned}P_{\text{SEP}} &= \Delta p + p_{\text{pressure}} \\ &= 0.58 + 20.67 \\ &= 21.25 \text{ psia} \quad (\text{SEPARATOR PRESSURE})\end{aligned}$$

$$T_{\text{SEP}} = 230.8 \text{ OF} \quad \nu_f = 0.016856 \quad \nu_g = 19.294$$

$$Re = \frac{VD}{\nu} \quad (\text{REYNOLDS NUMBER})$$

$$\begin{aligned}&= \left[\frac{0.55 \times 19.294}{\pi (3/2 \times 12)^2} \right] \times \left[\frac{3/12}{0.023 \times 10^{-5} \times 19.294} \right] \\ &= 3.4 \times 10^5\end{aligned}$$

$$\frac{E}{D} = 0.0006 \quad \Rightarrow \quad f = 0.0185$$

$$\Delta p_{\text{PIPE}} = f \frac{L}{D} \frac{V^2}{2g} \quad (\text{PIPE } \Delta p \text{ LOSS})$$

$$\begin{aligned}&= 0.0185 \times \left(\frac{12'}{3/12} \right) \times \frac{(216)^2}{2 \times 32.2} \times \frac{1}{144 \times 19.294} \\ &= 0.23 \text{ psi}\end{aligned}$$

$$\Delta p_{\text{minor}} \quad \frac{L}{D} = 146 \quad K = 0.31 \quad 3''$$

$$\frac{L}{D} = 29 \quad K = 0.56 \quad 2''$$

$$\Delta p_{\text{minor } 3''} = \left(0.0185 \times 146 + 0.31 \right) \frac{(216)^2}{2 \times 32.2} \times \frac{1}{144 \times 19.294}$$

$$= 0.785 \text{ psi}$$

$$\Delta p_{\text{minor } 2''} = \left(0.019 \times 29 + 0.56 \right) \frac{(486)^2}{2 \times 32.2} \times \frac{1}{144 \times 19.294}$$

$$= 1.467 \text{ psi}$$

$$\Delta p_{\text{stream}} = \Delta p_{\text{pipe}} + \Delta p_{\text{minor}} \quad (\text{stream } \Delta p \text{ in 2 parts})$$

$$= 0.23 + 0.785 + 1.467$$

$$= 2.48 \text{ psi}$$

$$Re = \frac{VD}{\gamma} \quad (\text{REYNOLDS FLUID})$$

$$= \left[\frac{0.45 \text{ lb/sec} \times 0.016856}{\pi \left(\frac{3}{2} \times 12 \right)^2} \right] \left[\frac{3/12}{0.171 \times 10^{-3} \times 0.016856} \right]$$

$$= 1.34 \times 10^4$$

$$\frac{L}{D} = 0.0004 \Rightarrow f = 0.03 \quad (\text{PIPE CALCULATION})$$

FRICTION COEFFICIENT

$$\Delta p_{\text{pipe}} = f \frac{L}{D} \frac{V^2}{2g}$$

$$\Delta P_{\text{pipe}} = 0.03 \left(\frac{12}{3/12} \right) \times \frac{(0.155)^2}{2 \times 32.2} \times \frac{1}{144 \times 0.016856}$$

$$= 0.000221 \text{ psi}$$

$$\Delta P_{\text{minor}} \quad \frac{L}{D} = 146 \quad K = 0.31 \quad 3''$$

$$\frac{L}{D} = 29 \quad K = 0.56 \quad 2''$$

$$\Delta P_{\text{minor } 3''} = \left(f \frac{L}{D} + K \right) \frac{V^2}{2g}$$

$$= (0.03 \times 146 + 0.31) \frac{(0.155)^2}{2 \times 32.2} \times \frac{1}{144 \times 0.016856}$$

$$= 0.000720 \text{ psi}$$

$$\Delta P_{\text{minor } 2''} = \left(f \frac{L}{D} + K \right) \frac{V^2}{2g}$$

$$= (0.03 \times 29 + 0.56) \frac{(0.349)^2}{2 \times 32.2} \times \frac{1}{144 \times 0.016856}$$

$$= 0.001113 \text{ psi}$$

$$\Delta P_{\text{water}} = \Delta P_{\text{pipe}} + \Delta P_{\text{minor}}$$

$$= 0.000221 + 0.000720 + 0.001113$$

$$= 0.001833 \text{ psi}$$

$$X = \left[\frac{\Delta P_{\text{water}}}{\Delta P_{\text{steam}}} \right]^{\frac{1}{2}} = \left[\frac{0.001833}{2.48} \right]^{\frac{1}{2}} = 0.0272$$

$$\phi_g = 1.22 + 4.065 [X]^{0.6646}$$

$$\phi_g = 1.22 + 4.065 (0.0272)^{0.6646}$$

$$= 1.59$$

$$\phi_L = 1.22 + 4.065 \left[\frac{1}{x} \right]^{0.6646}$$

$$= 1.22 + 4.065 \left[\frac{1}{0.0272} \right]^{0.6646}$$

$$= 45.83$$

$$\Delta P = \phi_g^2 \Delta p_g$$

$$= (1.59)^2 \times 2.48 \text{ psi}$$

$$= 6.27 \text{ psi}$$

$$\Delta P = \phi_L^2 \Delta p_L$$

$$= (45.83)^2 \times 0.001833 \text{ psi}$$

$$= 3.85 \text{ psi}$$

LT Situation

$$P_{\text{PERMANENT}} = P_{\text{CHSE}} + \Delta P_{S-T} + \Delta P_{L-S}$$

$$= 20.67 \text{ psia} + 0.58 \text{ psi} + 6.27 \text{ psi}$$

$$= 27.52 \text{ psia}$$

$$\text{PRESSURE RATIO} = P_{in} / P_{out}$$

$$= 120 \text{ psia} / 27.52 \text{ psia}$$

$$= 4.36$$

$$\text{GIVEN } \dot{m} = 1 \text{ lb/sec} \quad (\text{ABOUT})$$

$$X_{\text{SEPARATOR}} = \frac{\dot{m}_s}{\dot{m}_T} = \frac{0.55 \text{ lb/sec}}{1 \text{ lb/sec}}$$

$$= 0.55$$

$$h_{\text{SEP}} = h_g + X_{\text{SEP}} h_{fg} \quad (e P_{\text{SEP}} = 21.3 \text{ psia})$$

$$= 199.2 + 0.55 \times 958.2$$

$$= 726.2 \text{ Btu/lb}$$

$$h_{\text{EXHAUST}} = h_g + X_{\text{EX}} h_{fg} \quad (e P_{\text{EX}} = 27.52 \text{ psia})$$

$$X_{\text{EX}} = \frac{h_{\text{SEP}} - h_{g \text{ EX}}}{h_{fg \text{ EX}}} \quad (\text{EXHAUST DRYNESS})$$

$$= \frac{726.2 - 213.3}{948.9}$$

$$= 0.54$$

BY TRIAL & ERROR USING GRAPHS (REF B)

$$X_1 \approx 0.50$$

$$\text{RPM} \approx 9000$$

$$\dot{W}_L \approx 24 \text{ kW}$$

MODEL EQUATIONS

SYSTEM MODEL
4/12/82 DA Packer

LYSHOLM

EXIT QUALITY

$$x_2 = A + B(P.R.)$$

(P.R. = PRESSURE RATIO)

$$B = A_B e^{(B_B \cdot x_1)} + C_B$$

(X₁ = INLET QUALITY)

$$A = A_A + B_A(x_1)$$

$$A_B = A_{AB} + B_{AB}(N)$$

(N = RPM)

$$B_B = A_{BB} + B_{BB}(\ln(N))$$

$$A_{AB} = 1.802 E-2$$

$$B_{AB} = 3.323 E-7$$

$$A_{BB} = 2.265$$

$$B_{BB} = -0.349$$

$$A_A = 5.683 E-2$$

$$B_A = 9.399 E-1$$

$$C_B = -1.000 E-2$$

MASS FLOW RATE

$$m = ((P.R. - 1) / (a + b \cdot P.R.)) + m_e$$

(m_e = THEORETICAL MASS FLOW)

$$m_e = (V_i \cdot N) / (V_f + V_{fs} \cdot x_1)$$

(V_i = INTAKE VOLUME)
(V_f = SPECIFIC VOLUME (20 psi))
(V_{fs} = Δfg SPECIFIC VOLUME)

$$b = a_b + b_b \cdot x_1$$

$$a = a_a + b_a \cdot x_1$$

$$a_a = a_{aa} e^{(b_{aa} \cdot N)}$$

$$b_a = a_{ba} + b_{ba} N$$

$$a_b = a_{ab} e^{(b_{ab} \cdot N)}$$

$$b_b = a_{bb} + b_{bb} \cdot N$$

$$a_{aa} = -5.403 E-2$$

$$b_{aa} = 3.638 E-4$$

$$a_{ba} = -4.031$$

$$b_{ba} = 3.375E-4$$

$$a_{ab} = 2.694E-1$$

$$b_{ab} = 1.968E-4$$

$$a_{bb} = 3.685$$

$$b_{bb} = -2.050E-4$$

$$V_i = 1.331E-4$$

$$f + 3 \text{ MIN/REC.SEC}$$

$$v_j = 1.789E-2$$

$$f + 3 / 16m \approx 120 \text{ P/C}$$

$$v_{fs} = 3.710$$

$$f + 3 / 16m \approx 120 \text{ P/C}$$

POWER

$$POWER = (PR - PR_1) / (a + b \times PR)$$

(PR₁ = MIN PR FOR TURN
OVER)

$$a = a_a + b_a N$$

$$b = a_b + b_b N + c_b N^2$$

$$PR_1 = e^{(b_c N)}$$

$$a_a = 6.093E-2$$

$$b_a = -9.934E-6$$

$$a_b = 6.651E-2$$

$$b_b = -9.590E-6$$

$$c_b = 5.113E-10$$

$$b_c = 1.144E-4$$

4/20/82

3

PIPING & SEPARATOR

$$\phi_G = 1.2 + 2.8(X)^{0.7947}$$

$$1.22 + 4.065(X)^{0.6646}$$

* (NOTE THESE EQUATIONS ARE CURVE FIT TO DATA)

$$\phi_L = 1.2 + 2.8\left(\frac{1}{X}\right)^{0.7947}$$

$$1.22 + 4.065\left(\frac{1}{X}\right)^{0.6646}$$

$$X = \left(\frac{\Delta P_L}{\Delta P_G}\right)^{\frac{1}{2}}$$

$$\Delta P_{LACT} = \phi_L^2 \Delta P_L$$

$$\Delta P_{GACT} = \phi_G^2 \Delta P_G$$

SOUNDS BE FINE
LOW QUALITY ΔP_G FOUND
TO BE CLOSE

$$f = 64/RE$$

$$6 \times 10^2 < RE < 4 \times 10^3$$

$$\frac{\epsilon}{D_e} = 0.0018(D_e)^{-1.0248}$$

 D_e (INCHES) HYDRAULIC DIAM
HOT FINISH, WROUGHT IRON

$$RE = \frac{VD}{\mu \times 12}$$

REYNOLDS NUMBER

$$K = \left[1 - \left(\frac{d_1}{d_2}\right)^2\right]^2$$

$$\mu_{STEAM} = 0.4772 \times 10^{-5} + 1.492 \times 10^{-8}(T) \quad \left\langle \frac{\text{lbm}}{\text{ft sec}} \right\rangle$$

$$T \text{ } \langle ^\circ\text{F} \rangle$$

$$100^\circ\text{F} < T < 390^\circ\text{F}$$

$$\mu_{WATER} = 0.0992(T)^{-1.1695}$$

$$\left\langle \frac{\text{lbm}}{\text{ft sec}} \right\rangle$$

$$\Delta P = \left(f \frac{L}{D} + K\right) \times \frac{V^2}{2g} \times \frac{1}{1.44 \times 12}$$

LOOK UP FOR f & $V_{f \&g}$

TURBINETURBINE LOSSES (KW)

$$L_d = 0.042 D^2 \rho (u/100)^{2.9}$$

DISK DRAG

$$L_b = 0.187 D \rho h^{1.25} (u/100)^{2.9}$$

BUCKET DRAG

U = PITCH SPEED (F/S) DW

D = PITCH DIAM C_N NOZZLE (F)P = STEAM DENSITY (LB/FT³)

h = MEAN BUCKET HEIGHT (IN)

GENERATOR LOSSES (KW) (NEED EXPRESSION FOR I)

$$L_{wf} = 1.023 \times 10^{-4} (\text{RPM})^{1.273}$$

$$L_{cr} = 0.1517 I^{2.3196}$$

WINDAGE & FRICTION

3200 → 3800 < 1% ERROR

CORE & RESISTANCE

1 → 2 < 1% 2.6 15% ERROR

TURBINE POWER

$$\dot{m} = 0.0165 A_2 P_1^{0.97} \text{ (lb/sec)}$$

$$A_2 = \pi \left(\frac{3}{8} \right)^2 \times 16 = 1.7671 \text{ in}^2$$

$$C_N = 0.96 - \left(\frac{V_s}{4430} \right)^{7.1}$$

(NOTE THESE EQUATIONS ARE CHANGE FOR THOSE ON PAGE 50)

$$C_m = 1 - \left(\frac{V}{4070} \right)^{8.42} *$$

$$C_w = (R_p)^{-0.042}$$

A₂ = CRITICAL AREAP₁ = SOURCE PRESSURE

NOZZLE LOSSES *

V_s = ISENTROPIC VELOCITYV_s = ISENTROPIC VELOCITY

SUPERSONIC BUCKET LOSSES

MOISTURE LOSSES

APPLY TO POWER

$$C_v = 0.91$$

$$C_v = 0.92$$

$$C_v = 0.93$$

1" BLIND

STATOR

2" PLATE

4/20/82

5

$$\begin{aligned}
 V_B &= \text{RPM} \times 2\pi r && (\text{BLADE SPEED}) \\
 &= 3600 \text{ RPM} \times 2 \times \pi \times 10.186'' / 60 \frac{\text{SEC}}{\text{MIN}} \times \frac{12''}{\text{ft}} \\
 &= 320 \text{ ft/SEC}
 \end{aligned}$$

$$V_R = \left[\left(V_1 \cos\left(\frac{\alpha}{\beta} \right) - V_B \right)^2 + \left(V_1 \sin\left(\frac{\alpha}{\beta} \right) \right)^2 \right]^{\frac{1}{2}}$$

$$\begin{aligned}
 \alpha &= 15^\circ && (\text{RELATIVE VELOCITY}) \\
 \beta &= 19^\circ \\
 \gamma &= 20^\circ \\
 \delta &= 20^\circ
 \end{aligned}$$

$$F = \dot{M} \Delta V \langle I_b \rangle \quad (\text{BLADE FORCE})$$

$$\begin{aligned}
 &= \dot{m} \left[\left(V_{IN \text{ BLADE DIRECTION}} - V_{OUT \widehat{BD}} \right)_{1^{st} \text{ STAGE}} \right. \\
 &\quad \left. \left(V_{IN \widehat{BD}} - V_{OUT \widehat{BD}} \right)_{2^{nd} \text{ STAGE}} \right]
 \end{aligned}$$

$$\begin{aligned}
 &= \dot{m} \left[(V_{1N} \cos 15^\circ - 320) + (V_{2R} \cos 19^\circ) \right. \\
 &\quad \left. + (V_{3N} \cos 20^\circ - 320) + (V_{4R} \cos 20^\circ) \right] \frac{1}{32.2}
 \end{aligned}$$

$$\dot{W}_L = F \times V_B \quad \langle \text{KW} \rangle$$

$$\begin{aligned}
 &= F \times \frac{19,200 \text{ ft/min}}{44,240 \frac{\text{ft lb}_f}{\text{min KW}}}
 \end{aligned}$$

REF : DIXON " FLUID MECHANICS, THERMODYNAMICS
OF TURBOMACHINERY, Pg 96-100.

$$\text{ACTUAL VELOCITY} = \text{THEORETICAL VELOCITY} \times C_{LOS}$$

C_{LOS} = SODERBERG'S CORRELATION

$$\zeta = \left[0.065 - 0.100 \times \frac{T}{B} \right] + 0.06 \times \left(\frac{E}{100} \right)^2$$

T = BLADE THICKNESS

B = CORD

E = FLUID DEFLECTION ANGLE

$$= 180 - (\theta_{EXIT} + \theta_{ETRAINS})$$

θ_{EXIT} = FIXED 19°

$$\theta_{ETRAINS} = \tan^{-1} \left[\frac{V_1 \sin 15^\circ}{V_1 \cos 15^\circ - V_B} \right]$$

$$\zeta' = \left[\left(\frac{l}{H} + 1 \right) (0.975 + 0.075 B/H) - 1 \right]$$

(CORRECTION FOR NONSTANDARD BLADE HEIGHT)

H = BLADE HEIGHT

$$\zeta'' = \zeta' \times \left[\frac{10.5}{Re} \right]^{1/4}$$

$$C_{LOS} = \zeta''$$

4/25/82

CALCULATION #3

DETERMINATION OF COMPRESSIBILITY :

(SPEED OF SOUND)

$$c = \sqrt{k g_c R T}$$

[pg 604 WILEY
CLASSICAL THERMO]

$$= \sqrt{1.135 \times 32.17 \frac{\text{lb}_m \cdot \text{ft}}{\text{lb}_m \cdot \text{sec}^2} \times 85.76 \frac{\text{ft} \cdot \text{lb}_f}{\text{lb}_m \cdot \text{R}} \times (212 + 460)^{\circ}\text{R}}$$

$$= 1451 \text{ ft/sec}$$

(MACH NUMBER)

$$M = \frac{V}{c}$$

(MACH NUMBER)

$$= \left[\frac{0.4 \text{ lb/sec} \times 26.8 \text{ ft}^3/\text{lb}}{\pi \left(\frac{3}{2} \times 12 \right)^2 \text{ ft}^2} \right] / 1451 \text{ ft/sec}$$

$$M = 0.15$$

CAN CONSIDER GAS INCOMPRESSIBLE IF

$$\frac{\Delta P}{P} = \frac{1}{2} M^2 \leq 0.05$$

[PS II SCHLICHTING
BOUNDARY LAYER THEORY]

$$M \leq 0.32$$

∴ CAN CONSIDER GAS INCOMPRESSIBLE

4/28/82

CALCULATION #4

WHAT IS FREE CONVECTION LOSS/TOTAL FLOW

$$\dot{q}_L = hA \Delta T \quad (\text{HEAT LOSS})$$

$$= 10 \text{ Btu/hr ft}^2 \times 12' \times 2\pi \frac{1.75'}{12} \times (220 - 70)^\circ\text{F}$$

$$= 16,493 \text{ Btu/hr}$$

$$\dot{q}_F = \dot{m} h_s = \dot{m} (h_{g1} + h_{g2}) \quad (\text{HEAT FLOW})$$

$$= 0.5 \text{ lb/sec} \times 3600 \frac{\text{SEC}}{\text{hr}} \times (218.9 + 0.25 \times 945.2)$$

$$= 821,160 \text{ Btu/hr}$$

$$\% = \frac{\dot{q}_L}{\dot{q}_F} \times 100 = \frac{1.6 \times 10^4}{8.2 \times 10^5} \times 100$$

$$= 2.01 \%$$

TRIVIAL

(RATIO OF
HEAT LOSS TO
HEAT FLOW)

CALCULATION #5

WHAT IS EFFECT OF 1 FT OF ELEVATION

$$\Delta Z = \frac{H}{v} = \frac{\text{HEAD}}{\text{SPECIFIC VOLUME}} \quad (\text{PSI})$$

$$1 \text{ ft} = \frac{1 \text{ ft}}{20.09 \frac{\text{ft}^3}{\text{lb}} \times 144} \quad \begin{array}{l} \text{e 20 psi} \\ \text{SAT STEAM} \end{array}$$

$$= \underline{0.000346 \text{ psi}} \quad (\text{TRIVIAL})$$

$$1 \text{ ft} = \frac{1 \text{ ft}}{0.016834 \times 144} \quad \begin{array}{l} \text{e 20 psi} \\ \text{SAT LIQ} \end{array}$$

$$= 0.413 \text{ psi} \quad (< 10\%)$$

$$\frac{1}{6} \text{ ft} = \left[\frac{2/12 \cdot \frac{1}{144}}{\left(\frac{1}{0.016834} + \frac{0.17}{2207} \right) - 1} \right] \quad \begin{array}{l} \text{e 20 psi} \\ x_1 = 17\% \end{array}$$

$$= \underline{0.07 \text{ psi}} \quad \text{TRIVIAL}$$

5/28/82

CALCULATION #6

DATA REDUCTION

$\dot{m}_{\text{STEAM}}$ = MASS FLOW STEAM
 $\dot{m}_{\text{WATER}}$ = MASS FLOW WATER
 $\dot{m}_{\text{TURBINE}}$ = MASS FLOW STEAM
 THROUGH TURBINE

INPUT MIXING PROPERTIES

$$h_{\text{STEAM}} = [\text{CHAN}^{\#9}, 15 \text{ psia}] = [279^{\circ}\text{F}, 15 \text{ psia}] = 1182 \text{ Btu/lb}$$

$$h_{\text{WATER}} = [\text{CHAN}^{\#7}, \text{SAT}] = [339^{\circ}\text{F}, \text{SAT LIQ}] = 310.2 \text{ Btu/lb}$$

$$\dot{m}_{\text{STEAM}} = [\text{CHAN}^{\#10} \times 0.1 \times 1.025 - 0.006] = [4.064 \times 0.1 \times 1.025 - 0.006] = 0.411 \text{ lb/sec}$$

$$\dot{m}_{\text{WATER}} = [\text{CHAN}^{\#11} \times 0.946 + 0.085] = [0.3611 \times 0.946 + 0.085] = 0.427 \text{ lb/sec}$$

$$X_{\text{QUALITY}} = \dot{m}_{\text{STEAM}} / \dot{m}_{\text{TURBINE}} = \frac{0.411}{0.411 + 0.427} = 0.49$$

$$h_{\text{MIXTURE}} = ((1-x)h_w + xh_s) = (1-0.49)310.2 + 0.49 \times 1182 = 738.11 \text{ Btu/lb}$$

INPUT TO LYSOLM ENGINE

$$h_1 = h_2 \text{ INPUT TO LYSOLM}$$

$$@ T_2 [\text{CHAN}^{\#5}, \text{SAT}] = (340^{\circ}\text{F}, \text{SAT})$$

$$X_{2 \text{ INPUT}} = \frac{h_1 - h_s}{h_{sg}} = \frac{738.11 - 311.13}{879.0} = 0.4858$$

$$P_2 = 117.992 \text{ psia} \quad (\text{FROM CHARTS})$$

$$\dot{S}_2 \text{ ENTROPY AT INPUT LTM} = [\text{CHAN \#5, SAT}] = [3400^\circ\text{F, SAT}]$$

$$= \dot{S}_g + x_2 \dot{S}_{gg} = 0.49 + 0.4858 \times 1.0992 = 1.024 \frac{\text{Btu}}{\text{lb} \cdot \text{R}}$$

EXHAUST TO LYSIDUM ENGINE (IDEAL - ISENTROPIC)

$$\dot{S}_3' = \dot{S}_2 \text{ IDEAL ISENTROPIC} = [\text{CHAN \#6, SAT}] = [2550^\circ\text{F, SAT}],$$

$$P_3 = 32.539 \text{ psia}$$

$$x_{3'} \text{ ISENTROPIC QUANTITY} = \frac{\dot{S}_3' - \dot{S}_g}{\dot{S}_{gg}} = \left[\frac{1.024 - 0.3747}{1.3182} \right] = 0.4926$$

$$h_{3'} \text{ ISENTROPIC ENTHALPY} = h_g + x_{3'} h_{gg} = [223.6 + 0.4926 \times 942.1]$$

$$= 687.7 \text{ Btu/lb}$$

$$\Delta h \text{ IDEAL ENTHALPY DROP} = (h_2 - h_3) = 738.1 - 687.7$$

$$= 50.41 \text{ Btu/lbm}$$

POWER FROM LYSIDUM

$$\text{Power Out} = [\text{CHAN \#12} \times 1000 \text{ in lb} \times \text{CHAN \#13} \times 10,000 \text{ RPM}$$

$$\times 1.183 \times 10^{-5} \frac{\text{KW}}{\text{RPM in lb}}) + (0.00012 \frac{\text{KW}}{\text{RPM}} \times \text{CHAN \#13} \times 10,000 \text{ RPM}) - 0.155 \text{ KW}] = 0.148 \frac{\text{Btu}}{\text{KW sec}}$$

$$= [(\text{MEASURED POWER}) + (\text{BELT LOSSES})$$

$$- (\text{oil, water, air power})]$$

$$\begin{aligned}
&= \left[(0.2182 \times 1000 \text{ in } \text{lb}_3 \times 0.5973 \times 10,000 \text{ RPM} \right. \\
&\quad \left. \times 1.183 \times 10^{-5} \frac{\text{KW}}{\text{RPM in}^3} \right) \\
&\quad + \left(0.00012 \frac{\text{KW}}{\text{RPM}} \times 0.5973 \times 10,000 \right) \\
&\quad \left. - 0.155 \text{ KW} \right] 0.948 \frac{\text{Btu}}{\text{SEC KW}} \\
&= 15.15 \text{ Btu/SEC} \quad (15.98 \text{ KW})
\end{aligned}$$

$$\begin{aligned}
\eta_{\text{L45100cm}} &= \frac{\text{POWER OUT}}{\Delta h_{\text{ISOTHERMAL}} \times \dot{m}} \\
&= \left[\frac{15.15 \text{ Btu/SEC}}{50.41 \frac{\text{Btu}}{\text{lb}} \times 0.8372 \text{ lb/SEC}} \right] \\
&= 36\% \quad (\text{ENGINE})
\end{aligned}$$

$$P_2/P_3 = \left[\frac{117,992}{32,539} \right] = 3.63 \quad \text{PRESSURE RATIO}$$

TURBINE POWER

$$h_{\text{LINE}} = [\text{CHAN} \# 20, 15 \text{ psia}] = [220^\circ\text{F}, 15 \text{ psia}] = 1154.3 \frac{\text{Btu}}{\text{lbm}}$$

$$\begin{aligned}
x_{\text{LINE}} &= [\text{CHAN} \# 24, \text{SAT}] = [255^\circ\text{F}, \text{SAT}] = \frac{h_{\text{LINE}} - h_f}{h_{fg}} \\
&= \left[\frac{1154.3 - 223.6}{942.1} \right] = 0.988
\end{aligned}$$

$$P_{\text{LINE}} = 32.54 \text{ psia}$$

$$\begin{aligned} S_{LINE} &= S_g + x_{LINE} S_{fg} = 0.3747 + 0.988 \times 1.3182 \\ &= 1.6770 \text{ Btu/lb} \cdot \text{R} \end{aligned}$$

$$\begin{aligned} x_{TURBINE} &= \left[\text{CHAN}^{\#} 25, \text{SAT OR CHAN}^{\#} 31 \times 0.2 + 14.7 \text{ psia, SAT} \right] \\ &= [236 \text{ OF } 2, \text{SAT}] \quad [23.216 \text{ psia}] \\ &= \frac{h_{LINE} - h_g}{h_{fg}} = \frac{1154.3 - 204.29}{954.8} = 0.995 \end{aligned}$$

$$\begin{aligned} S_{TURBINE} &= S_g + x_{TURBINE} S_{fg} = 0.3473 + 0.995 \times 1.3725 \\ &= 1.7129 \text{ Btu/lb} \cdot \text{R} \end{aligned}$$

$$\begin{aligned} S_{EXHAUST} &= S_{TURBINE} + [\text{CHAN}^{\#} 26, \text{SAT}] \\ &= 1.7129 \frac{\text{Btu}}{\text{lb} \cdot \text{R}} + [138^{\circ}\text{F, SAT}] \end{aligned}$$

$$x_{EXHAUST} = \frac{S_E - S_g}{S_{fg}} = \frac{1.7129 - 0.1959}{1.7129} = 0.886$$

$$\begin{aligned} S'_{EXHAUST} &= S'_{LINE} + [\text{CHAN}^{\#} 26, \text{SAT}] \\ &= 1.6770 + [138^{\circ}\text{F, SAT}] \end{aligned}$$

$$x'_{EXHAUST} = \frac{S'_{LINE} - S_g}{S_{fg}} = \frac{1.6770 - 0.1959}{1.7129} = 0.873$$

$$\begin{aligned} h_{EXHAUST LINE'} &= h_g + x_{E'} h_{fg} + [\text{CHAN}^{\#} 26, \text{SAT}] \\ &= [138^{\circ}\text{F, SAT}] \end{aligned}$$

$$= 106.42 + 0.873 \times 1014.9 = 992.34 \text{ Btu/lb}$$

$$\begin{aligned}
 h_{EXHAUST\ C\&EST} &= h_g + x_E h_{g, 9} [C\&AN \# 26, SAT] \\
 &= [138^\circ F, SAT] \\
 &= 106.42 + 0.886 \times 1014.9 = 1005.62 \frac{Btu}{lbm}
 \end{aligned}$$

$$\begin{aligned}
 \text{POWER TURBINE \# 2} &= [C\&AN \# 35 \times 10 \frac{KW}{VOLT} + WINDAGE: \\
 &\quad C\&AN \# 38 + I^2 R \ C\&AN \# 37] \\
 &= 18.46 KW + 3.42 KW + 0.36 KW \\
 &= 22.24 KW
 \end{aligned}$$

$$\begin{aligned}
 \dot{m}_{TURBINE} &= [C\&AN \# 32 \times 0.8] \times 0.984 + 0.016 \\
 &= 0.614 \times 0.8 \times 0.984 + 0.016 \\
 &= 0.4993 \text{ Btu/sec}
 \end{aligned}$$

$$\begin{aligned}
 \eta_{LINE} &= \frac{\text{POWER TURBINE OUT}}{[h_{LINE} - h_{2'}] \dot{m}_{TURBINE}} \\
 &= \frac{22.24 KW \times 0.948 \text{ Btu/sec KW}}{[1154.3 - 992.339] 0.4993} \\
 &= 26.1\% \quad (\text{TURNS INCLUDES THROTTLE})
 \end{aligned}$$

$$\begin{aligned}
 h_{C\&EST} &= [C\&AN \# 25, SAT] = [236^\circ F, SAT] \\
 &= h_g + x h_{g, 9} \\
 &= 204.4 + 0.995 \times 954.8 \\
 &= 1154.4 \text{ Btu/lbm} \quad [\text{THROTTLE LINE GOOD ASSUMPTION}]
 \end{aligned}$$

$$\begin{aligned}
 \eta_{\text{CHESST}} &= \frac{\text{Power Turbine Out}}{\{h_{\text{CHESST}} - h_E\} \dot{m}_{\text{turbine}}} \\
 &= \frac{22.24 \text{ kW} \times 0.946 \text{ Btu/sec kW}}{\{1154.4 - 1005.62\} 0.4993 \frac{\text{Btu}}{\text{sec}}} \\
 &= 28.36 \%
 \end{aligned}$$

TURBINE POWER OUT = 22.24 kW

PRESSURE CHESST = 23.216 psia

$\dot{m}_{\text{STEAM TURBINE}}$ = 0.4993 lb/sec

LYSHOLM POWER = 15.98 kW

LYSHOLM PRESSURE RATIO = 3.63

4/29/82

1502

CALCULATION #7LYSOLM EXPANSION FOR SUB ATMOSPHERIC
EXPANSION $\dot{m}$ = TOTAL MASS FLOW RATE η_L = LYSOLM EFFICIENCY η_{pump} = PUMP EFFICIENCY X = QUALITY INDEX η_T = TURBINE EFFICIENCY

$$NET\ WORK_L = \dot{m} (h_1 - h_2) \eta_L - \frac{\dot{m} \Delta P}{\rho} \frac{1}{\eta_{pump}}$$

$$\dot{m} = 11 \text{ lb/sec} \times X, \quad / \text{ PUMP STREAM TURBINE } \dot{m}$$

$$\eta_L = ? \quad / 70\% \eta_{turbine}$$

$$\eta_{pump} = 70\%$$

$$p_1 = 14.7 \text{ psia}$$

$$= \left[\dot{m} (h_1 - h_2) \eta_T - \frac{\dot{m} \Delta P}{\rho} \frac{1}{\eta_{pump}} \right] X$$

$$NW_L = NW_T \quad (\text{NET WORK})$$

$$(h_1 - h_2) \eta_L - \frac{\Delta P}{\rho} \frac{1}{\eta_{pump}} = \left[(h_1 - h_2) \eta_T - \frac{\Delta P}{\rho} \frac{1}{\eta_{pump}} \right] X$$

$$\eta_L = \left[(h_1' - h_2') \eta_T X + (1-X) \frac{\Delta P}{\rho} \frac{1}{\eta_{pump}} \right] / (h_1 - h_2)$$

For $X = 50\%$

$P_1 = 14.7$

$\eta_T = 0.70$

$P_2 = 5$

$\eta_P = 0.70$

$\dot{m} = 1 \text{ lb/sec}$

$r = 60.28 \quad (60.05)$

$$\eta_L = \frac{\left[(1150.5 - 1076.7) 0.7 \times 0.50 + \frac{(1 - 0.5)(9.7) \times 0.0012854}{60.28/144 \times 0.70} \right]}{[665.35 - 627.45]}$$

= 68.1%

P_2	X	h'_1	x'_2	h'_2	h_1	x_2	h_2	η_L
5	0.25	1150.5	0.95	1076.7	422.75	0.27	407.84	65.0
5	0.50	1150.5	0.95	1076.7	665.35	0.49	625.45	68.1
5	0.75	1150.5	0.95	1076.7	907.95	0.72	852.07	69.4
10	0.25	1150.5	0.98	1123.1	422.75	0.26	415.68	68.0
10	0.50	1150.5	0.98	1123.1	665.35	0.50	651.49	69.3
10	0.75	1150.5	0.98	1123.1	907.95	0.74	887.30	69.7

4/30/82

CALCULATION #B

EXHAUST STEAM QUALITY

$$\dot{m}_s = X_2 \times \dot{m}$$

(LYNCHMAN MODEL EXHAUST QUALITY)
(TIMES MASS FLOW RATE)

$$= [A + B \cdot P.R.] \times \left[\frac{(P.R. - 1)}{(a + b \cdot P.R.)} + W_1 \right]$$

FOR 9000 RPM : $X_1 = 0.5, 0.25, 0.17$
 $P.R. = 5.31, 4.1$

X_1	A	B	a	b	$M \pm$	PR 4.1	PR 5.31
						$\dot{m}_s$	$\dot{m}_s$
		0.0033		2.505		(1.0114)	(1.0188)
0.5	0.527	-1.924			0.640	0.549	0.555
		0.00673		2.044		(1.729)	(1.7366)
0.25	0.291	-1.676			1.267	0.551	0.567
		0.00799		1.896		(2.3488)	(2.3557)
0.17	0.217	-1.596			1.847	0.587	0.611
		-0.00157		3.424		(0.588)	(0.5945)
1	0.997	-2.421			0.321	0.582	0.588

CALCULATION #9

DOES $\dot{m}_{S_2}$ INCREASE AS V_L INCREASES FOR
CONSTANTS X_1 & N

$$\dot{m}_{S_2} = \dot{m} \times X_2$$

$$\dot{m}_{S_2} = \left[(P.R. - 1) / (a + b \times P.R.) + m_e \right] [A + B \times P.R.]$$

$$P.R. = \frac{C_3 V_L}{V_T} + C_2 \quad \text{[CLOSE TO O.P.E.]}$$

$$m_e = V_L C_1$$

$$\dot{m}_{S_2} = \left[\left((-V_L C_3' + C_2 - 1) / a + b \times (V_L C_3' + C_2) \right) + V_L C_1 \right] \times [A + B (-V_L C_3' + C_2)]$$

$$\frac{d\dot{m}_{S_2}}{dV_L} = X_2 \frac{d\dot{m}_{S_2}}{dV_L} + \dot{m} \frac{dX_2}{dV_L} \quad \text{[0.0]} +$$

$$\left\{ \frac{[-C_3' [a + b(-C_3' V_L + C_2)] - b C_3' [-V_L C_3' + C_2 - 1]]}{[a + b(-C_3' V_L + C_2)]^2} + C_1 \right\}$$

$$+ X_2 [A + B(-V_L C_3' + C_2)]$$

$$+ \left\{ [-B C_3'] \left[\left(\frac{-V_L C_3' + C_2 - 1}{a + b(-V_L C_3' + C_2)} \right) + V_L C_1 \right] \right\}$$

$$= \left[\left(\frac{-(a + b \cdot P.R.) C_3' - b(P.R. - 1)}{a + b(P.R.)^2} + C_1 \right) (0.55) \right] + [-B \times 1] C_3'$$

$$N = 900$$

$$P.R. = 5.51$$

$$X_1 = 0.5$$

$$= \left[\left(\frac{-(a+b)C_3'}{(a+bPR)^2} + C_1 \right) 0.55 \right] + \left[C_3' B \times 1 \right]$$

$$B = \left[1.802E-2 + 3.323E-7 \times 9000 \right] \times e^{[(2.265 - 0.349 \ln 9000) 0.5]} - 0.001$$

$$= 2.04 \times 10^{-3}$$

$$C_1 = \frac{9000}{(1.789E-2 + 3710 \times 0.5)}$$

$$= 4805.4$$

BY EXAMINING TERMS, $\frac{dm_{s2}}{dV_i}$ IS + FOR V_i GOING UP. THIS MEANS MORE STEAM BEING DELIVERED TO TURBINE.