# APPENDIX C

# TURBOMACHINERY SYSTEM ANALYSIS FOR STRATIFIED CHARGE ROTARY ENGINE CRITICAL TECHNOLOGY/ENABLEMENT

SUNDSTRAND AEROSPACE

☐ ELECTRIC POWER SYSTEMS

☐ MECHANICAL SYSTEMS

☐ FLUID SYSTEMS

☐ POWER SYSTEMS

☐ TECHNICAL SERVICES OPERATIONS

☐ MANUFACTURING OPERATIONS

☒ RESEARCH

REPORT NO. AER 3485

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# ENGINEERING REPORT

## Turbomachinery System Analysis For Stratified Charge Rotary Engine Critical Technology Enablement

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## ABSTRACT

Analytical studies of turbomachinery systems to enhance the performance of stratified charge rotary engines were conducted under subcontract to John Deere Technologies International, Inc. (JDTI). JDTI was under contract with NASA Lewis Research Center (NAS3-25945) to demonstrate critical technologies for a general aviation aircraft engine in the Stratified Charge Rotary Engine Critical Technologies Enablement (SCRECTE) program. The design objective is an engine suitable for a high performance twin-engine light aircraft. To obtain adequate power density in high altitude cruise, as well as for stratified charge operation, turbocharging is mandatory. The turbomachinery analytical studies were conducted to determine the optimum system configuration. The Sundstrand effort included assembly of component sizing and performance prediction models into a system analysis model for evaluation of candidate systems. Evaluation criteria were developed, and an optimization routine used to find the preferred configurations. The best turbomachinery configuration included a compounding turbine, which converts some of the exhaust energy to shaft power, as well as a turbocharger having two compressor stages driven by a single-stage turbine. The intake charge was intercooled between compressor stages and between the second compressor and the engine intake port. Intake manifold pressure control was effected by exhaust wastegate operation.

KEYWORDS

Keywords or phrases for computerized technical information data base: SCRECTE, rotary engine turbocharger, turbochargers, general aviation, and turbocompound engine.

## 1.0 SUMMARY

The NASA Lewis Research Center contracted with John Deere Technologies International, Inc. (JDTI), under the Stratified Charge Rotary Engine Critical Technologies Enablement (SCRECTE) program, to demonstrate a competitive general aviation engine. The demonstrator was based on previous JDTI designs but required significant development in fuel economy and power density at cruise altitude. The engine specification required 40 percent thermal efficiency (brake specific fuel consumption, BSFC, of 0.34 lbs/hp/hr on Jet-A fuel) at cruise conditions and specified full power to be available from sea level to 25,000 feet altitude and 75 percent power for cruise up to 33,000 feet. In order to meet the power requirement at altitude, intake port air density needed to be as high as previously demonstrated at sea level. The compressor could not use shaft power, because this would increase BSFC, which needed to be lowered. The source of energy available to improve the BSFC and provide intake pressurization is the engine exhaust. The flow rates, pressures, and temperatures involved strongly suggest the use of turbomachines for these processes. JDTI employed Sundstrand to analyze the turbomachinery system for the engine.

Sundstrand originally planned, in a multiphase program, to provide system analysis, including component sizing and performance predictions; to design the turbomachinery and ancillary equipment; to procure, assemble, and test the turbomachinery system; and, finally, to support engine testing with the turbomachinery system. The program was subsequently descoped due to alterations in the contractual agreements between JDTI and NASA, and the Sundstrand effort was terminated before the design hardware effort was begun.

The system analysis process leading to system configuration selection, essentially complete at program termination, consisted of several tasks.

- Creation of candidate system configurations. Many suggestions were compiled from various sources, including Sundstrand, JDTI, and consultants.

- Assembly of component models. Each element of the system, including turbines, compressors, heat exchangers, ducts, and gearboxes, needed to be characterized in terms of weight, performance, and volume. Most of the component models were available or easily adapted from existing Sundstrand models.

- Define the parameters of interest in assessing the candidate systems, in order to be able to select the "best" configuration.

- Encode a computer program, including an optimization algorithm, to find the best implementation of each of the candidate system configurations, to estimate the characteristics of the components, and to calculate the system selection parameters.

The design/sizing analysis was essentially complete at program termination, and a recommended system configuration was selected. Some additional detailed analysis was needed to confirm estimated exhaust pulse recovery benefits, which were expected to provide significant improvements in exhaust energy recovery and, therefore, engine BSFC.

The selected turbomachinery system configuration incorporates a turbocharger consisting of two compressor stages and one turbine stage on a single shaft, intercooling after each compressor stage, and a compounding turbine upstream of the turbocharger (at the engine exhaust). The system is estimated to weigh 206.50 lbm, to impose a net ram drag of 5.05 lbf on the aircraft, and to produce 49.4 hp of shaft power through turbocompounding (increasing engine output). The following sections of the report describe the above activities in more detail.

## 2.0 ANALYTICAL MODELS

In order to rapidly evaluate a number of candidate system configurations, computer models were constructed to estimate and optimize characteristics of components. These were combined into a system thermodynamic analysis computer program which balances component functions and computes system parameters. An optimization function, called equivalent power, was defined to enable the "best" system to be selected (the optimization function derivations are given in Section 3.3).

### 2.1 System Model

A typical turbomachinery system for a rotary engine is shown in Figure 1. This system, like all the systems evaluated, employs various turbine and compressor stages, heat exchangers for rejection of charge air heat of compression to ambient air, charge air lines and ram air ducts. A gearbox is employed for the power turbine options. Each of these component types is modelled by unique component sizing subroutines which predict component size and weight. Most of the routines perform heat transfer and thermodynamic analysis and some use empirically derived sizing relationships. The line sizing subroutine was adapted for this contract; the remaining subroutines existed before the contract began and are Sundstrand proprietary.

The system analysis computer program can be adapted to model various system configurations by altering input flags. A system schematic with various configuration options is shown in Figure 2. The program calculates system state points (temperature, pressure and flow rate) at the numbered locations, based on the input design conditions, constraints and the current values of the optimization parameters (defined in Section 3.4) and calls the various component sizing models to size the components based on the calculated state points. The equivalent power function is evaluated for the resultant values of engine exit pressure, power turbine power, incurred drag and fixed weight. An optimization routine is included in the system analysis computer program (flow chart provided in Figure 3). The optimization routine varies the system optimization parameters until the equivalent power is maximized.

### 2.2 Turbines and Compressors

The turbomachinery models incorporated in the system model use specific speed curves and empirical relations to determine wheel diameters which are then used with scaling functions to estimate component weight. For the turbomachines which include more than one wheel on a shaft (i.e. one turbocharging spool), the shaft speed is selected such that

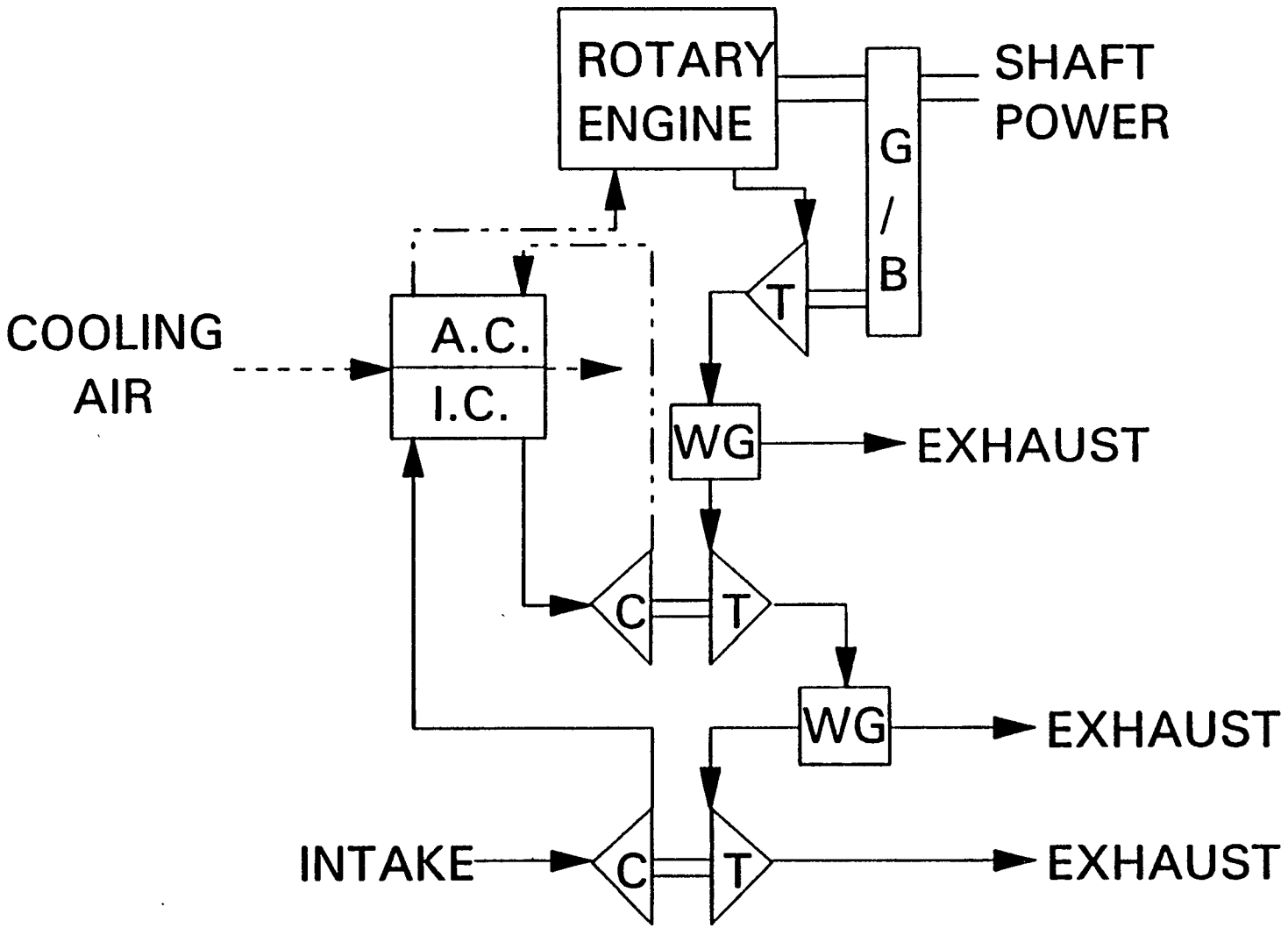



FIGURE 1. TURBOCOMPOUND ENGINE

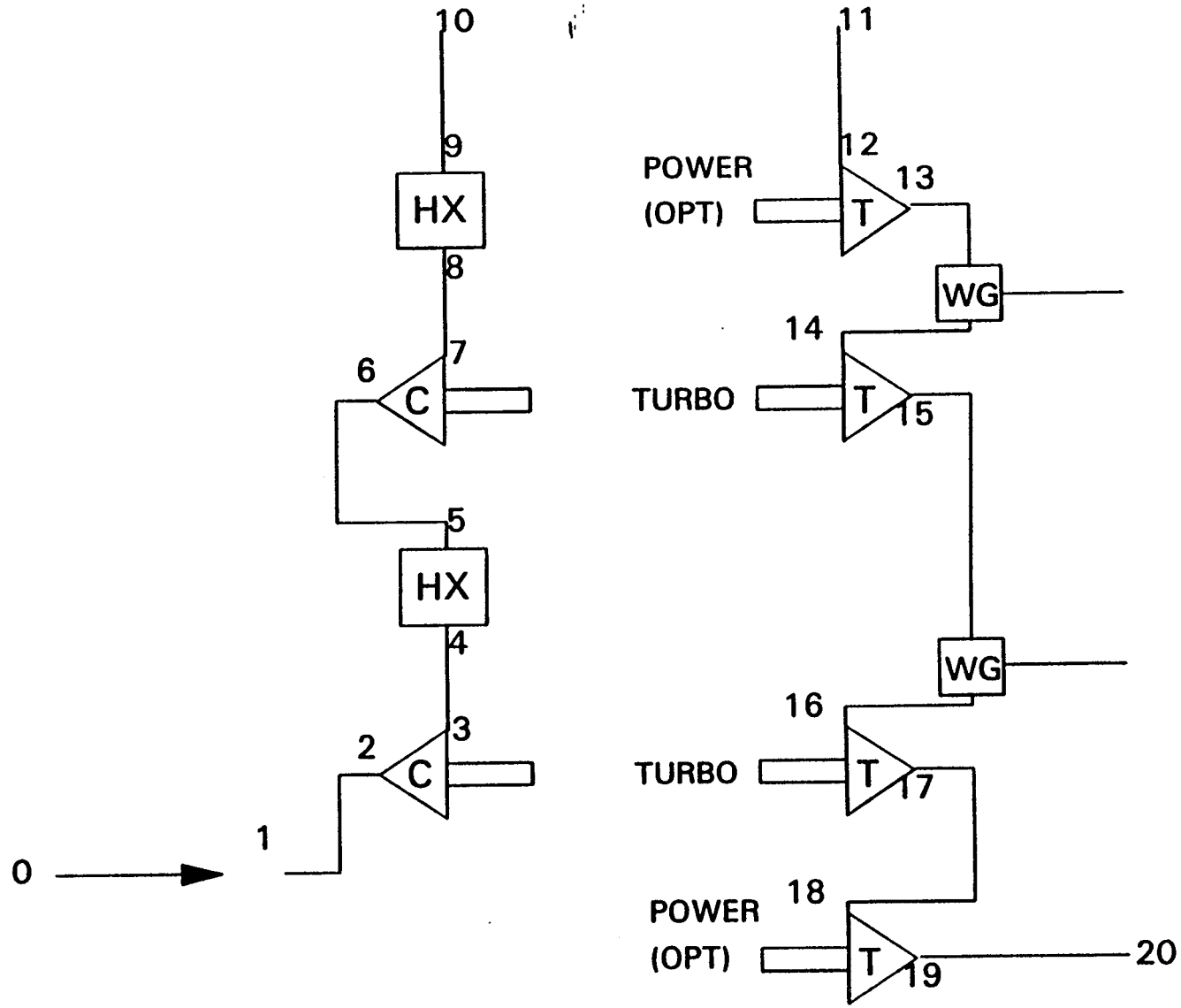



FIGURE 2 TURBOMACHINERY SYSTEM STATE POINTS

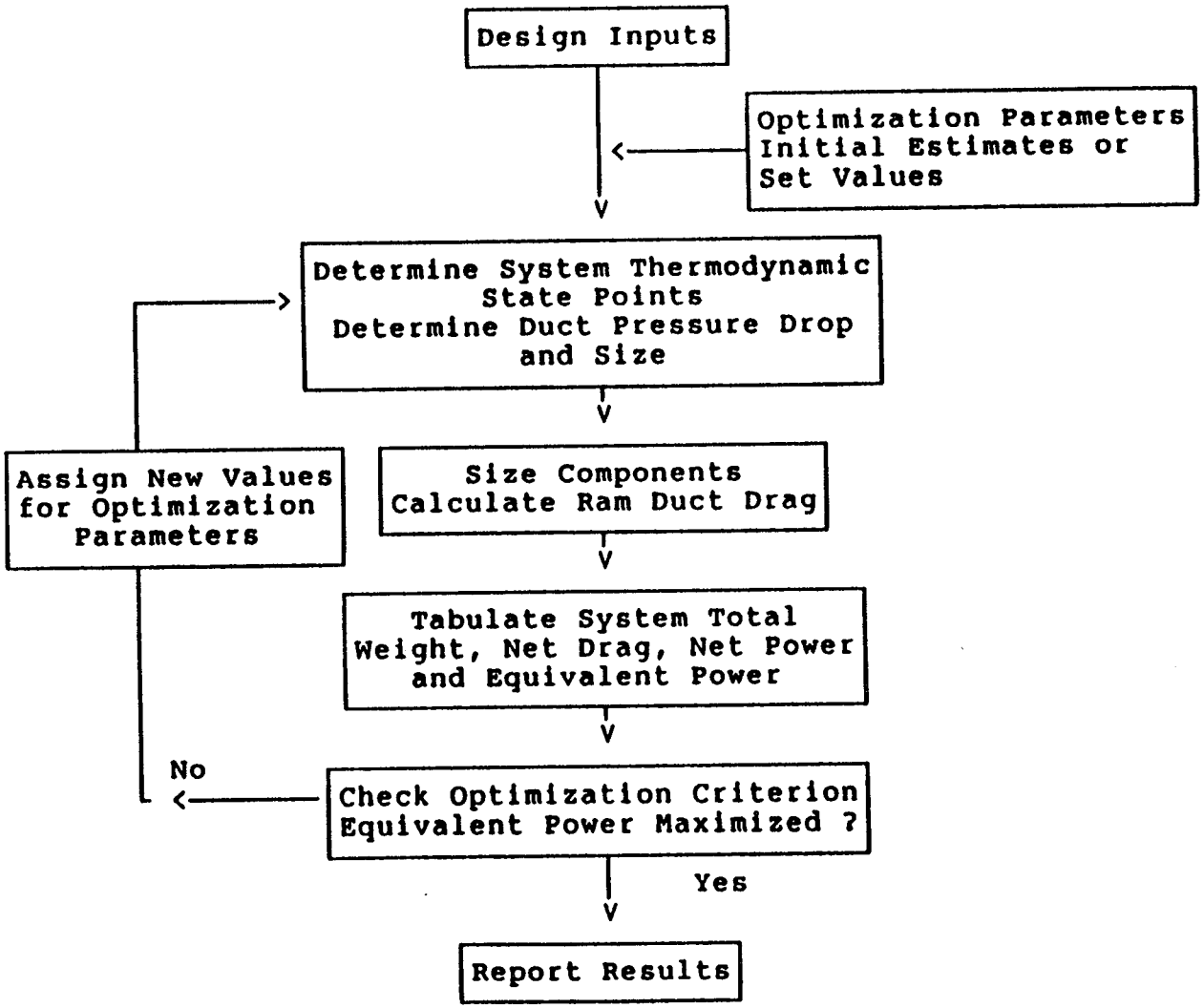



**FIGURE 3. SYSTEM ANALYSIS PROGRAM FLOW CHART**

all of the wheels on that shaft operate at near-optimum specific speeds. The acceptable specific speed ranges are set such that efficiencies range between 0.7 and 0.85. For turbomachines with only one wheel on a shaft (i.e. a power turbine on a gearbox), the shaft speed is determined such that the specific speed is optimal for maximum efficiency.

Compressor and turbines sizes are determined by somewhat different techniques. For compressors, the tip speed, determined by the head coefficient, and shaft speed are used to determine wheel diameter. For turbines, a specific speed/specific diameter correlation for peak efficiencies is used to determine the optimum specific diameter; the wheel diameter is calculated from the specific diameter. In either case, the wheel diameter is used in an empirical correlation to determine wheel and housing weight. The weight correlation has been calibrated using turbocharger data supplied by Beech Aircraft.

Turbine and compressor designs generated by the system models have been checked using more detailed Sundstrand proprietary computer programs to assure reasonable accuracy. The results of these check runs provided the efficiency predictions input to the system program. Turbomachinery inlet conditions and isentropic head rise are input to the design routine, which then calculates shaft speed, turbomachine weight, and volume.

## 2.3 Heat Exchangers

The heat exchangers are modelled as crossflow plate fin devices. Alternating layers of fin stock and parting plates make up the heat exchanger core as illustrated in Figure 4. Aluminum fins with aluminum parting plates are used for heat exchangers with service temperature below 350°F while nickel fins with stainless steel parting plates are used for service temperatures above 350°F. Typical values of fin height, spacing and thickness, parting plate thickness, etc. are assigned to each side of the heat exchanger based on the whether the fluid is liquid or gas. The required heat exchanger UA is determined by the effectiveness-NTU method. An iterative approach (outlined in Kays and London [1]) where the hot and cold side mass fluxes are iterated until the desired pressure drop on each side is met, is used to determine heat exchanger geometry. Plate fin correlations for friction factor and Colburn J factor are used to determine friction factor and heat transfer coefficient. The design routine takes hot and cold side inlet and exit temperature, pressure drop and flow rate as input and calculates heat exchanger weight, volume and face area.

## 2.4 Ducts

The duct model determines duct pressure drop, heat loss, and weight for a specified inlet Mach number. The duct area is determined which yields the specified inlet Mach number for the required duct flow rate. The duct pressure drop is calculated based on a friction factor correlation and the determined duct diameter. Insulation properties, local ambient conditions, and duct diameter are used to determine the duct heat loss. Finally, weight and volume are determined based on the duct diameter and wall thickness and the insulation properties. Duct inlet conditions, duct and insulation length, thickness, density and conductivity, and local ambient conditions are input to the routine and duct pressure drop, temperature drop, weight and volume are calculated.

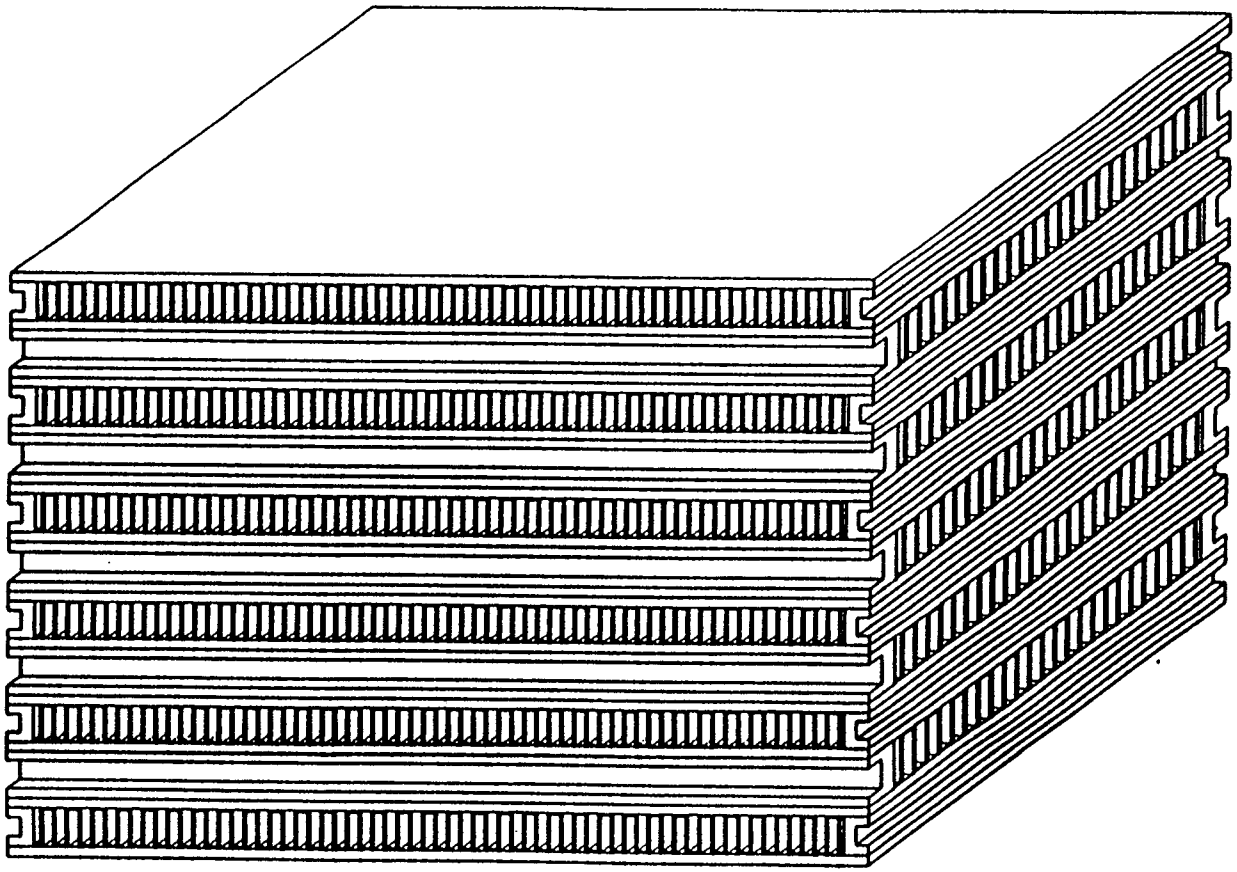


**FIGURE 4. CROSSFLOW HEAT EXCHANGER CORE**

A ram air circuit model calculates the duct losses and drag incurred by the ram air ducts. This routine tracks total pressure and accounts for frictional losses through an assumed duct geometry. The duct configuration, shown in Figure 5, includes a flush inlet, an inlet diffuser, a heat exchanger, a 90° bend section leading to a fan, the fan, a section of duct leading to an exit nozzle, and an exit nozzle. The SCRECTE system does not require a ram air fan, so the fan and the upstream 90° bend duct are deleted from the ram air duct analysis by a flag in the subroutine. Friction factor correlations and diffuser and nozzle loss correlations are used to model the pressure drops in the circuit. Ambient conditions, heat exchanger flow area and pressure drop, ram air flow rate, and fan pressure rise are input to the model and duct drag and fan power are calculated. If the inlet area is not specified (and usually it should not be) it is determined such that the inlet Mach number is 76 percent of the free stream Mach number. Inlet Mach numbers lower than this could cause boundary layer separation, causing flow instability and large losses at the inlet. Due to the many possible ram air duct installations the weight of the ram air duct is not estimated.

## 2.5 Exhaust Pulses

All of the component models, as well as the system model, predict performance and size of components on the basis of steady state flow conditions. It is well known, however, that the intermittent flow in the exhaust of an internal combustion causes pressure and flow pulses which greatly alter the distribution of energy with time and affect turbomachine size and performance. In order to assess these effects, a model of the exhaust system was created to calculate correction factors to be used in the system model. The transient model integrates the engine exhaust port and turbine inlet flows to determine the rate of change of mass in the exhaust duct and the resulting exhaust duct pressure. Wave dynamics have been ignored. Figure 6 shows some sample calculations of one exhaust cycle of one rotor; the variations shown are due to changes in the exhaust duct exit pressure and in exhaust duct length (volume). The taller pulses contain more energy, indicating the need for low volume exhaust systems. The pressures shown are not high enough to affect engine internal flow processes. The pulse modelling was not completed, but clearly indicates that care should be taken in the sizing of the exhaust system to preserve the power potential of the pulses. Preliminary results also indicate that the pulsed flow can contain 50 to 100 percent more power than a steady flow, and that up to 50 percent larger turbine and duct flow areas are required than for steady flow. Turbine average efficiency is lower than for steady flow by a few percentage points, but this is a small effect compared to the extra available energy, which greatly increases turbine power.

## 2.6 Gearboxes

Gearbox weight is determined from a correlation of weight versus power. Weights were estimated for several gearbox designs over a range of power turbine powers and speeds to derive this correlation. An allowance for unsteady shaft dynamics was included by sizing all gearbox elements for three times the rated power. Engine rotor accelerations and decelerations, due to intermittent combustion, are amplified through the power turbine gear train and can result in large loads and/or shaft deflections. It is recognized that a much more rigorous shaft dynamics analysis is needed for actual gearbox design.

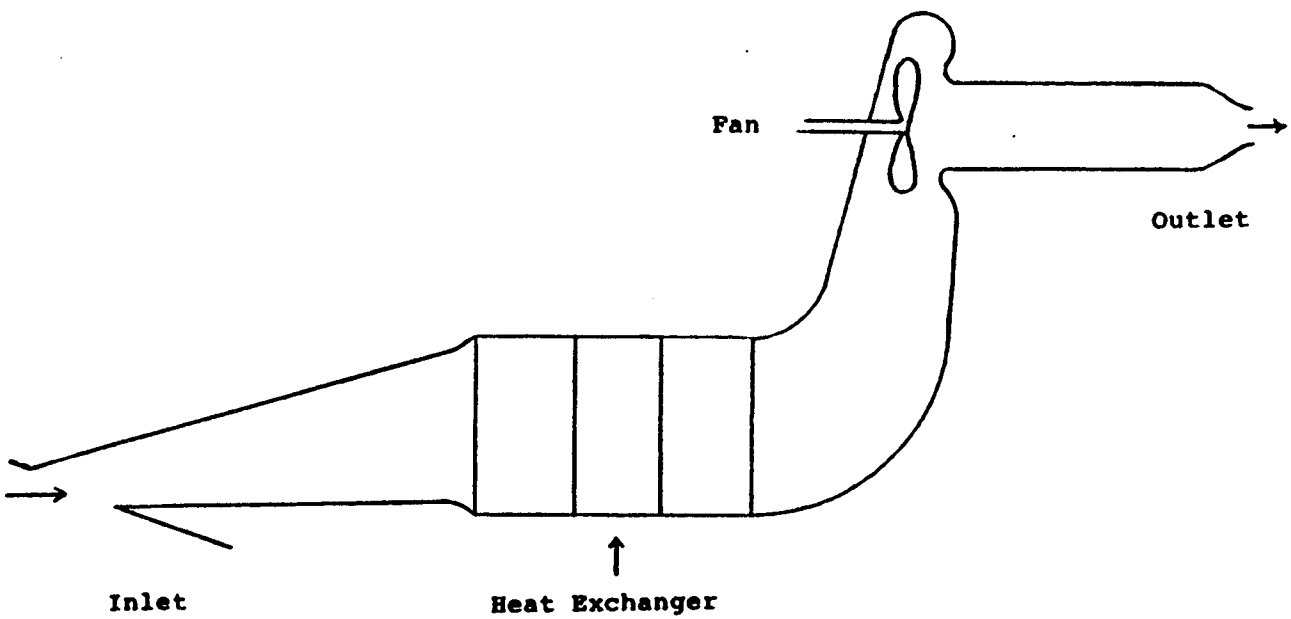



**FIGURE 5. RAM AIR DUCT SYSTEM**

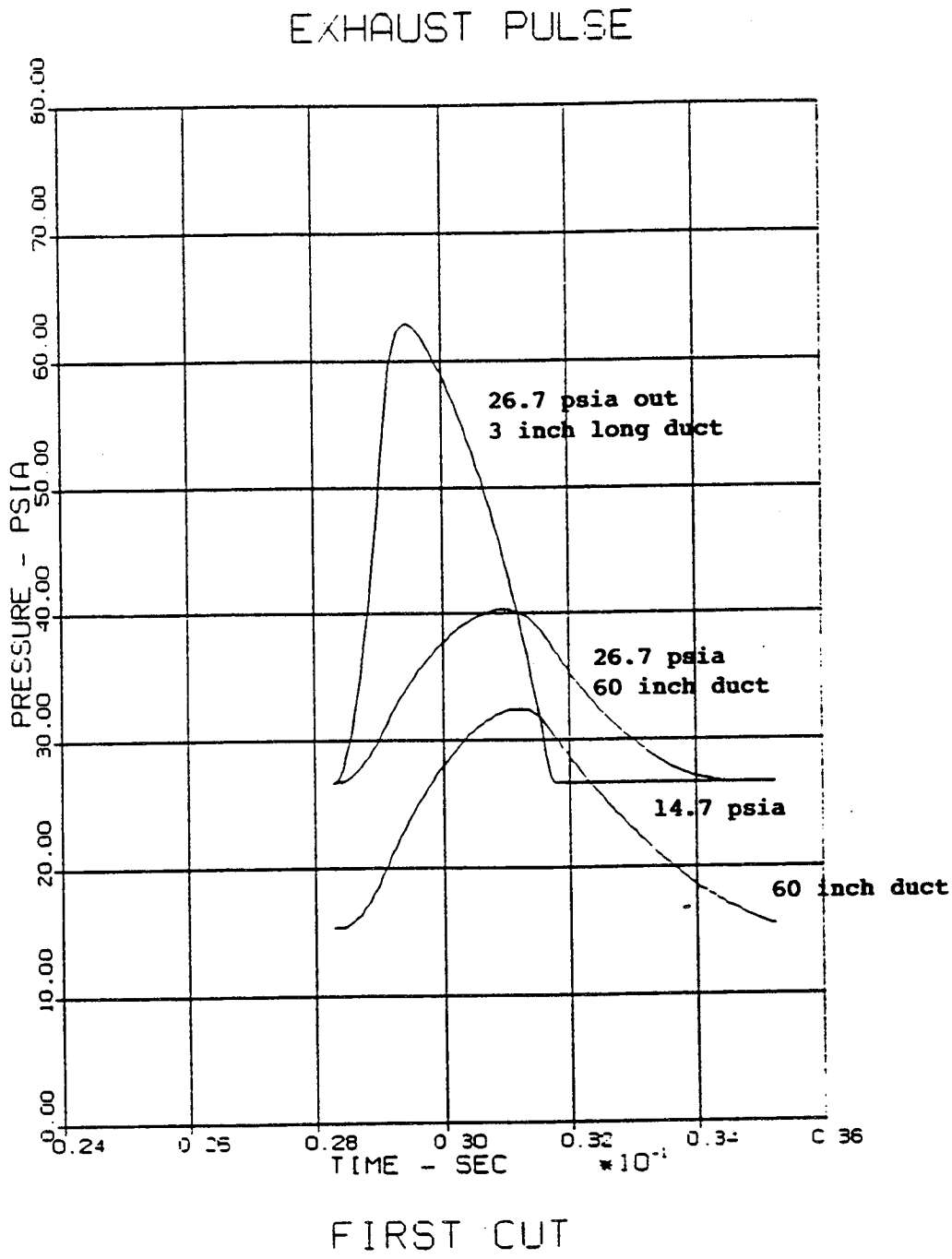



**FIGURE 6. EXHAUST PULSES**

## 3.0 SYSTEM EVALUATION

The functional requirements for the turbomachinery system are described in Section 3.1. These form the boundary conditions for analysis of any proposed system. The comparative evaluation of candidate turbomachinery systems required several steps subsequent to development of the system analysis computer program. Initially, an array of candidate system configurations was identified by brainstorming and extraction from publications or other experienced sources on turbocharging. Some sorting of these schematics was done based on operational considerations, the rest using the system analysis program. The candidate systems are discussed in Section 3.2. Secondly, it became clear that there was no clear definition of what constituted the "best" turbomachinery system; significant effort was expended in defining and encoding the optimization parameters. These are discussed in Section 3.3. Finally, the system analysis/optimization computer program was used to calculate comparative values of the system criteria for the most promising system configurations. Results are given in Section 3.4.

### 3.1 Design Conditions

A list of design conditions for 25 kft climb, which constrain each turbocharger system design, is given in Table 1. These values are either taken or calculated from information given to Sundstrand by John Deere and Beech Aircraft. Table 2 lists the assumed design constraints which depend on installation.

The total pressure and temperature of the flow entering the heat exchanger cooling circuit ram air scoop are affected by the engine propeller. An analysis at the propeller, based on propeller information supplied by Beech Aircraft, has been performed to determine these conditions . The thrust developed by the propeller and the power required to supply this thrust can be expressed as:

$$T = W * (V_2 - V_1) / g_c$$
$$W = \rho_1 * V_1 * A_1$$
$$A_1 = A_p$$
$$T = \rho_1 * V_1 * A_p * (V_2 - V_1) / g_c$$

$$HP = T * V_1 / \eta_p / 550$$
$$T = HP * \eta_p * 550 / V_1$$

Definitions:
- $T$ - thrust
- $W$ - air flow rate
- $V_2$ - air velocity downstream of propeller
- $V_1$ - air velocity upstream of propeller (true air speed)
- $\rho_1$ - upstream density
- $A_p$ - propeller diameter
- $HP$ - engine power
- $\eta_p$ - propeller efficiency

So $V_2$ can be expressed as:

$$V_2 = V_1 + HP*\eta_p*550*g_c / (\rho_1*V_1^2* A_p)$$

Table 2
Assumed Design Constraints

| | | |
|---|---|---|
| Charge Air Ducts | | |
| Compression side | | |
| Length (ft) | | 5.0 |
| Material | | Al |
| Wall thickness (in) | | 0.03125 |
| Density (lbm/ft$^3$) | | 172.8 |
| Conductivity (Btu/hr ft °F) | | 110.0 |
| Expansion side | | |
| Length (ft) | | 3.0 |
| Material | | Steel |
| Wall thickness (in) | | 0.03125 |
| Density (lbm/ft$^3$) | | 499.5 |
| Conductivity (Btu/hr ft °F) | | 8.67 |
| Insulation | | |
| Density (lbm/ft$^3$) | | 11.9 |
| Conductivity (BTU/hr ft °F) | | 0.10 |
| Inlet and Exit | | |
| Inlet diffuser half angle (deg) | | 10.0 |
| Exit nozzle half angle (deg) | | 10.0 |
| Ratio of nozzle exit to freestream momentum | 1.0 | |
| Heat loss to surroundings | | |
| Heat transfer coefficient (Btu/hr ft$^2$ °F) | 2.0 | |
| Ram air heat exchanger cooling duct | | |
| Inlet diffuser half angle (deg) | | 10.0 |
| Angle of exit flow relative to free stream | | 0.0 |

Table 1
Design Conditions
25000 ft climb

| | |
|---|---|
| Ambient | |
| Pressure (psia) | 5.46 |
| Temperature (°F) | 6.71 |
| Mach number | 0.267 |
| Downstream of Propeller | |
| Pressure (psia) | 5.84 |
| Total Temperature (°F) | 16.14 |
| Mach number | 0.311 |
| Engine air requirements | |
| Inlet temperature (°F) | 140.0 |
| Inlet flow rate (lbm/min) | 70.55 |
| Inlet Pressure (psia) | 34.9 |
| Exit temperature (°F) | 1154.0 |
| Exit flow rate (lbm/min) | 72.84 |
| Engine parameters | |
| Propeller efficiency | 0.83 |
| Displaced volume ($in^3$) | 80.0 |
| Engine Speed (rpm) | 8500.0 |
| Exit pressure for 400 HP (psia) | 30.6 |
| Aircraft lift over drag | 11.2 |

The total temperature and Mach number downstream of the propeller

$T_{T2} = T_{T1} + HP*550/778/(c_p*W)$ $T_{T1}$ - upstream (ambient)
total temperature
$M_2 = V_2 / (\gamma*R*T_{T2} - (\gamma-1)/2*V_2^2)^{.5}$ $T_{T2}$ - downstream total temperature
$c_p$ - specific heat
$M_2$ -downstream Mach number
$\gamma$ - specific heat ratio
R - gas constant

The total pressure downstream of the propeller is then:

$P_{T2} = P_{S1}*(1 + (\gamma-1)/2*M_2^2)^{\gamma/(\gamma-1)}$ $P_{T2}$ - downstream total temperature
$P_{S1}$ - upstream (ambient) static temperature

The heat transfer coefficient between the hot expansion side ducts and the surrounding air in the engine cowl has been determined based on a simple free convection correlation. The correlation applies to free convection around a heated circular duct.

## 3.2 Candidate Systems

Several types and arrangements of equipment have been considered for the turbomachinery system. These are shown and discussed below.

The turbocharger system consists of six characteristic functions, which are illustrated in Figure 7. The turbocharger extracts power from the exhaust stream and uses it to compress the intake charge. The intercooler (I.C.) and aftercooler (A.C.) reduce the temperature of the compressed air to increase the intake charge density. A power turbine can be used to extract additional exhaust energy for use as engine output shaft power. A gearbox is needed to connect the power turbine to the engine shaft. The exhaust wastegates bypass exhaust gas around the turbocharger to limit intake pressurization. Ducts provide the interface connections among the system components and between the system and the engine and airframe.

The system shown as an example in Figure 7 is a simple, thermodynamically complete turbomachinery system for a turbocompounded, turbocharged engine system. Many different candidate system arrangements can be devised from this system by varying the number of compression stages, the number of turbine stages, the order in which the exhaust gases pass through the turbines, and the presence or absence of the power turbine which provides the turbocompounding. A few of these possible configurations are shown in Figures 8 through 10.

Figure 8 shows three system configurations without turbocompounding. The first is composed of two conventional turbochargers (multiple spool system), each having a single compressor stage and a single turbine stage, with intercooling after each compressor stage. Intake manifold pressure is limited by sequential operation of the wastegates, starting with the low pressure spool. The second configuration combines the two compressors on a single shaft driven by a single turbine. This multistage turbocharger system promises to

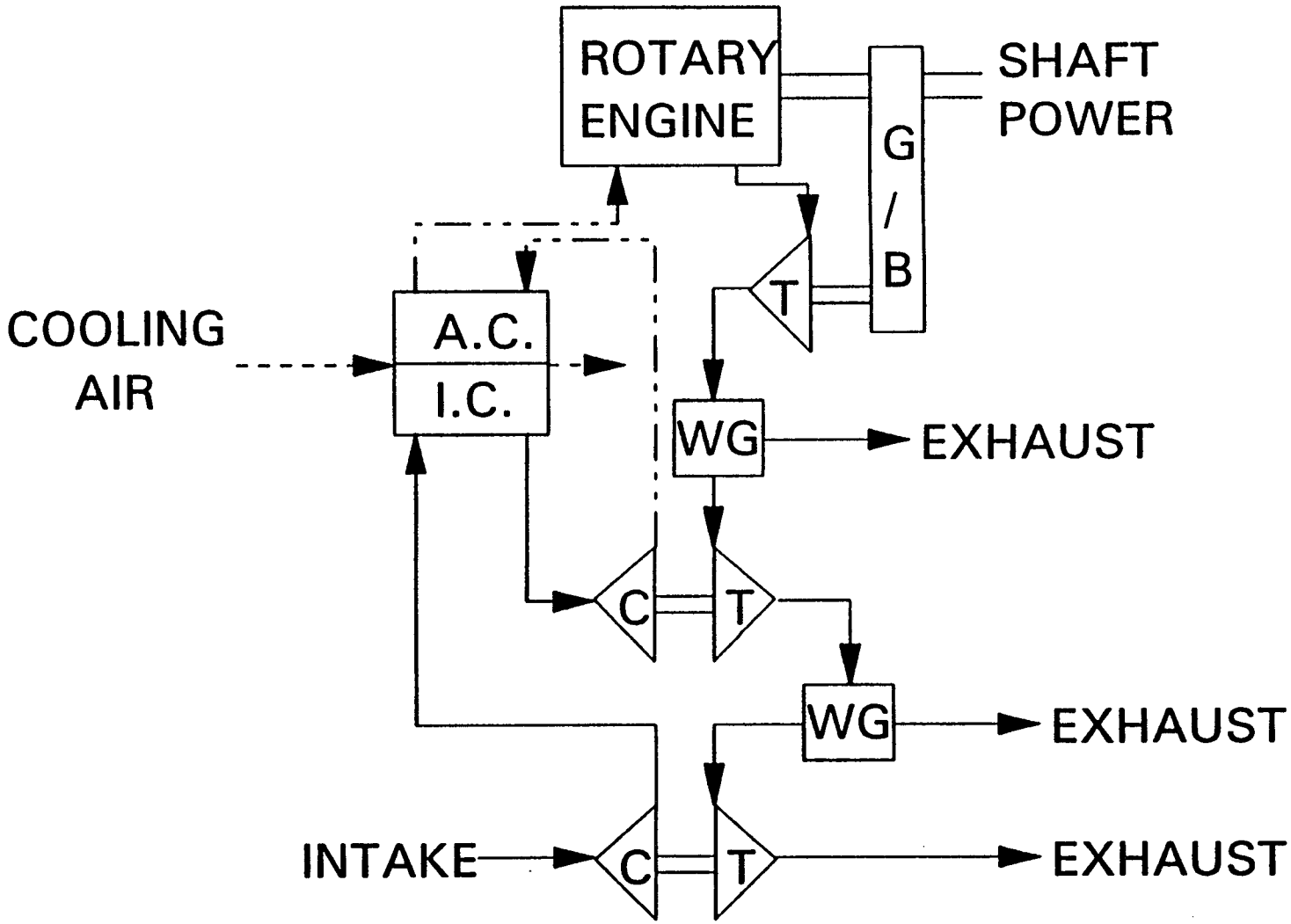
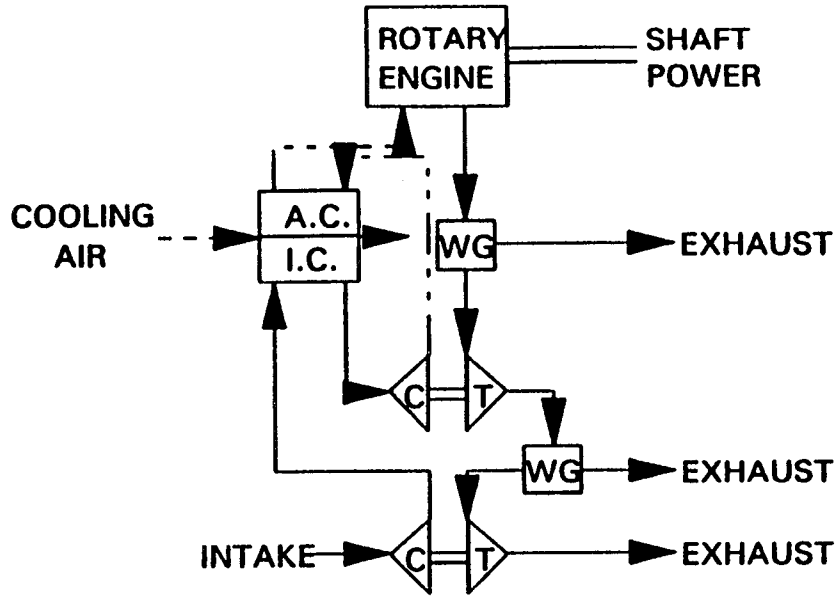
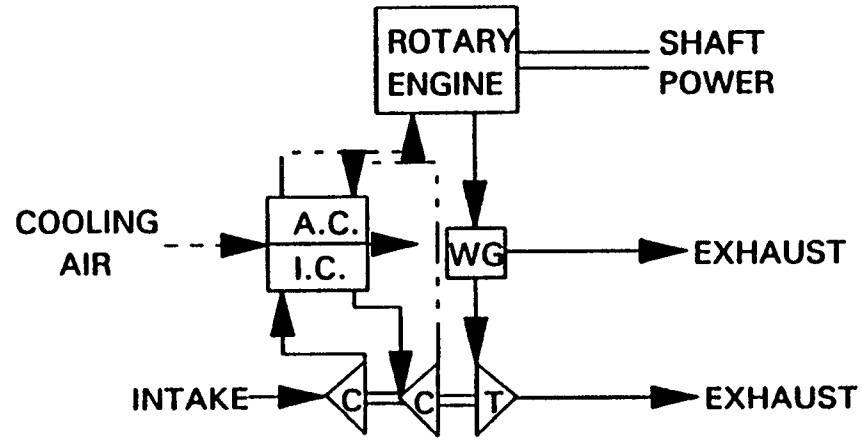


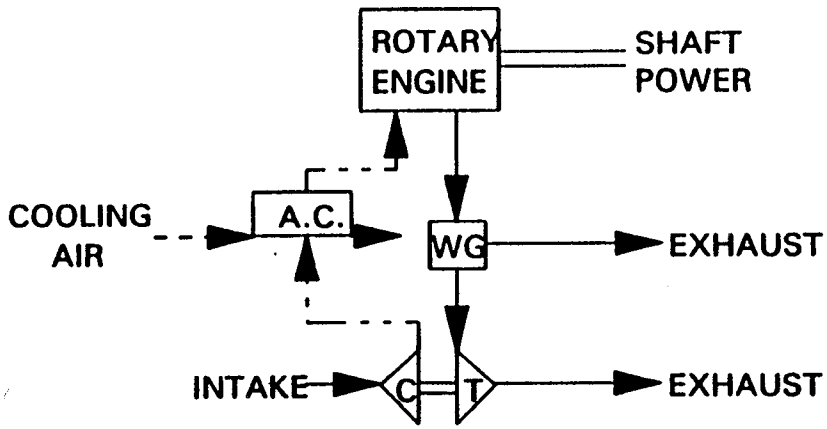

FIGURE 7. TURBOCOMPOUND ENGINE




MULTIPLE SPOOL TURBOCHARGER




MULTIPLE STAGE TURBOCHARGER




SINGLE STAGE TURBOCHARGER

FIGURE 8. CANDIDATE SYSTEMS (NO COMPOUNDING)

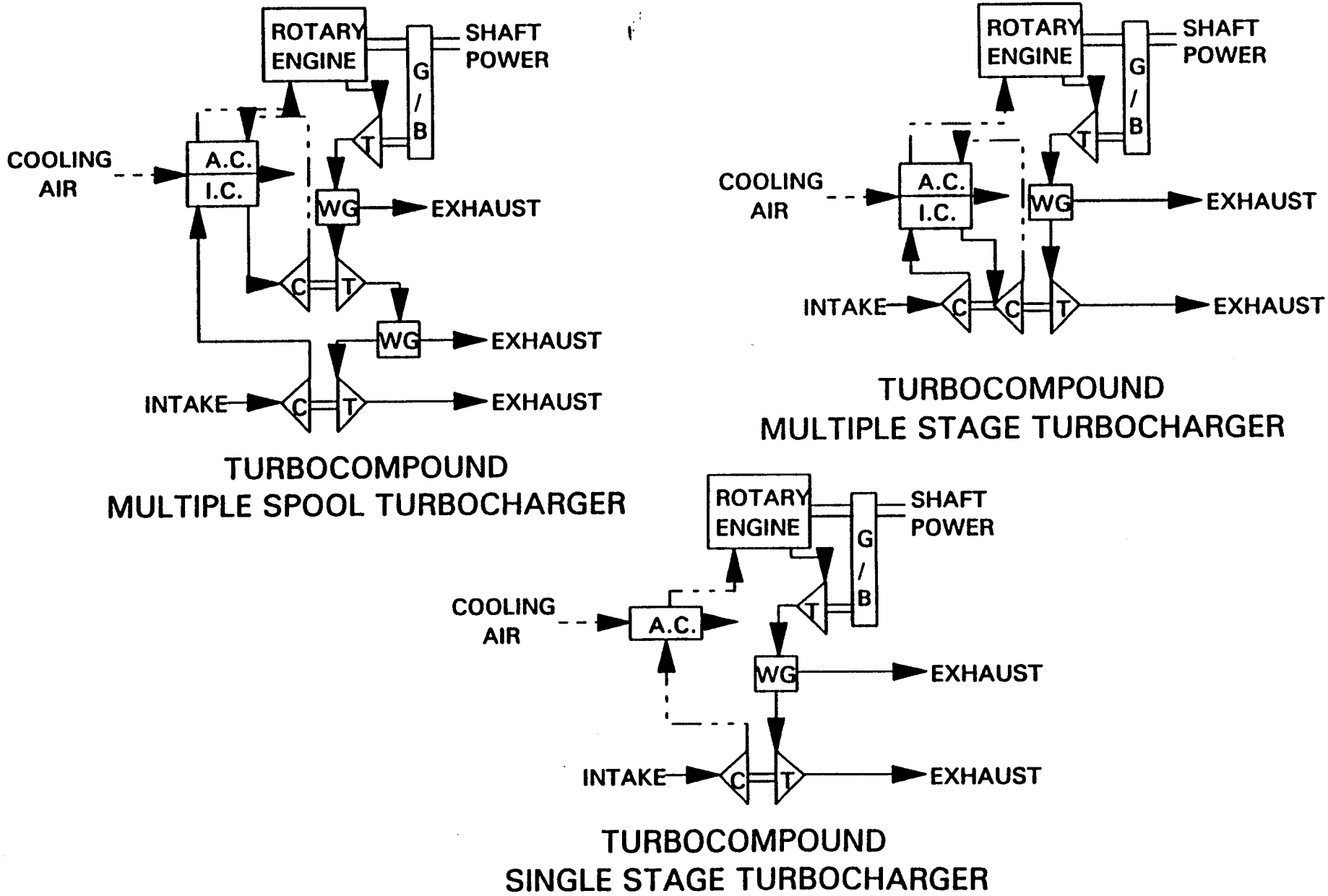
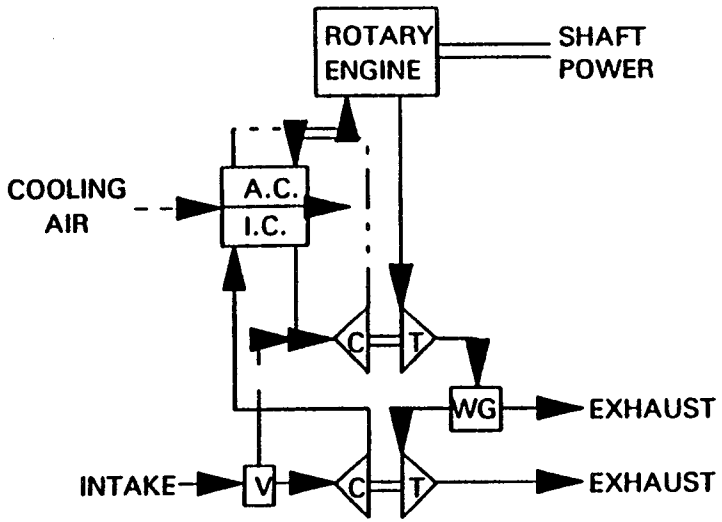
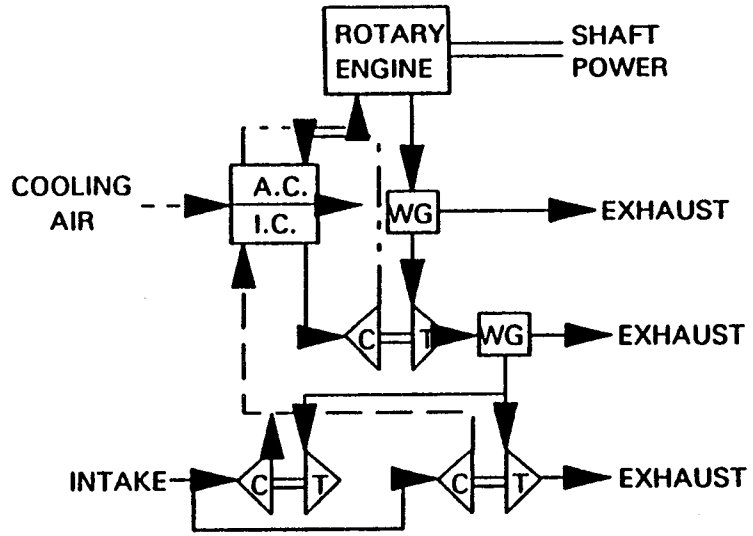


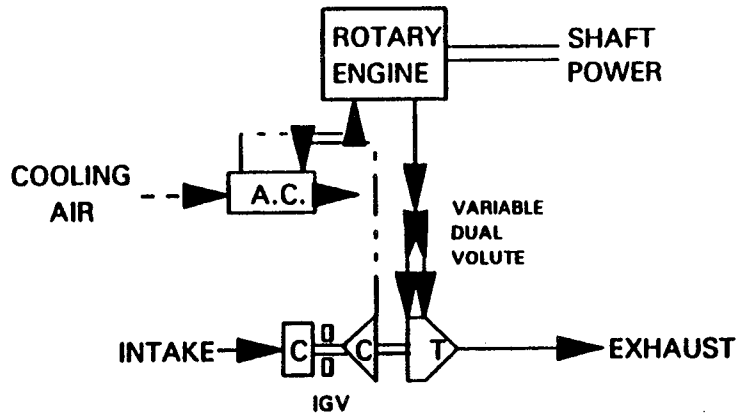

FIGURE 9. TURBOCOMPOUND CANDIDATE SYSTEMS



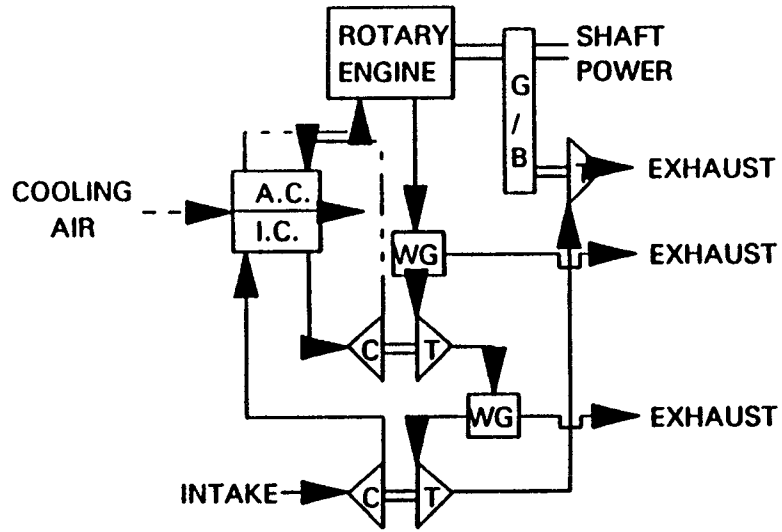

REDIV ALTERNATE 1




3 EQUAL TURBOCHARGERS




REDIV ALTERNATE 2




ALTERNATE TURBOCOMPOUND ENGINE

# FIGURE 10. ALTERNATE CANDIDATE SYSTEMS

be more compact than the multispool system, while delivering similar thermodynamic performance. The third system achieves the required compression in a single stage. The single stage turbocharger appears to provide the simplest system, but was determined to have thermodynamic disadvantages and to be a more risky approach.

The three systems in Figure 9 are similar to those in the previous figure, except that each has a power turbine in the exhaust stream preceding the turbocharger(s). This turbine provides the "turbocompounding" effect, extracting excess power from the exhaust to add to the engine output shaft power. The analysis shows that there is significant gas power available for turbocompounding, and that this function may be necessary to achieve the engine BSFC goal.

In Figure 10 are shown four other schematics which were proposed but discarded for various reasons. The first configuration is like the basic multiple spool configuration, but uses an additional valve to control the compressor flow split. This causes additional compressor matching problems without adding any performance to the system. The second schematic, REDIV Alternate 2, suggests using a variable geometry turbine instead of wastegate control and an axial/centrifugal compressor. This is thermodynamically equivalent to the multistage turbocharger at design operation, which is the only operating condition analyzed to date. Some operational (off-design) advantages may accrue from using variable geometry instead of wastegate control. Controls would be more complex, and development would be necessary. The three equal turbochargers were proposed as a potential approach for constructing a ground test article to demonstrate thermodynamic performance. The additional pieces of hardware and plumbing would make it impractical for aircraft installation. The alternate turbocompound schematic illustrates the potential for positioning the power turbine last in the exhaust, instead of first. This is suggested in the literature as a technique for improving engine throttle response, because it tends to maintain turbocharger turbine pressure ratios at a more constant level. However, it also eliminates the increase in turbocharger output with increasing altitude, which is an absolute must for the aircraft engine.

## 3.3 Optimization Criteria

A mathematical optimization process, such as included in the system analysis, requires an objective function which smoothly varies with the independent parameters of the problem. The NASA contract for this engine development set goals for power, weight, and fuel consumption of the entire system; however, the allotments for the turbomachinery were not known, nor was there any guidance given for tradeoffs. In order to proceed with the system optimization, it was necessary to develop optimization functions.

The turbocharger system imposes three penalties on the airframe and engine. These are the system fixed weight, the drag incurred by the ram air cooled heat exchangers, and the back pressure imposed at the engine exhaust. The formulated penalty function relates the relative impact on the airframe and engine of these three factors. The optimization parameters are varied (or optimized) by an optimization algorithm to minimize this penalty function and thus maximize the benefits of the turbomachinery system to the aircraft.

The approach taken to formulate the penalty function recognizes that the purpose of the rotary engine and its turbomachinery system is to provide propulsive power to the airframe. Benefits and penalties to the aircraft performance which are generated by the turbomachinery system can be accounted as a combination of thrusts and drags (or the equivalent powers). Thus, the ram drag of the system, drag due to lift incurred by system weight, engine power changes due to exhaust pressure changes, and power turbine contributions can all be combined into a single expression for net output power due to the turbomachinery system. (The "best" turbomachinery, by this definition, is the one which maximizes installed thrust-minus-drag.)

Drag:

The system fixed weight and incurred drag can be combined into the net drag imposed on the airframe by the turbomachinery system. The weight results in a drag penalty on the airframe due to additional lift required; this can be expressed by dividing the weight by the lift over drag (LOD) ratio. The net drag function takes the form:

$$\text{net drag} = \text{incurred drag} + \text{fixed weight/LOD}$$

Power:

The engine exhaust pressure can be converted into a net power change from the baseline power, produced by the engine and power turbine pair. The pumping loop contribution to engine power can be expressed as:

$$Pw_{pl} = \Delta V \times N \times (Pe_{in} - Pe_{out})$$

$Pw_{pl}$ - pumping loop power
$N$ - engine speed
$\Delta V$ - swept volume
$Pe_{in}$ - engine inlet pressure
$Pe_{out}$ - engine exhaust pressure

The power at some reference engine exhaust pressure has been given by the engine manufacturer. The pumping loop power portion of total engine power at this reference exhaust pressure ($Pe_{out,ref}$) is given by:

$$Pw_{pl,ref} = \Delta V \times N \times (Pe_{in} - Pe_{out,ref})$$

Assuming that a change in the exit pressure effects only the pumping loop power, the change in the engine power (which is just the change in pumping loop power) for this exit pressure is given by:

$$\Delta Pw = Pw_{pl} - Pw_{pl,ref}$$
$$\Delta Pw = \Delta V \times N \times (Pe_{out,ref} - Pe_{out})$$

The net power change from the engine power baseline also includes the power turbine contribution. The net power change is given by:

$$\Delta Pw_{net} = \Delta Pw + Pw_{pt}$$

$Pw_{pt}$ - power to propeller shaft from power turbine

Equivalent Power:

The net effect of the turbomachinery system on engine shaft power available for propulsion is obtained by converting the net drag to a (negative) thrust power and adding to the net power. The "equivalent power" optimization function for the turbocharging system has the form:

$$Pw_{eq} = \Delta Pw_{net} + V_{free} \times \text{net drag} / \eta_{prop}$$

$V_{free}$ - free stream velocity
$\eta_{prop}$ - propeller efficiency

or,

$$Pw_{eq} = \Delta V \times N \times (Pe_{out,ref} - Pe_{out}) + Pw_{pt} - V_{free} \times (\text{incurred drag} + \text{fixed weight/LOD})$$

## 3.4 Parametric Evaluations

Optimization Parameters:

Nine parameters have been identified which affect the optimization function, as well as weight, drag, and engine exhaust pressure. Some affect only one or two of these. The optimization parameters are listed below along with a description of the penalty factors they affect.

| | | |
|---|---|---|
| 1. | First stage compression ratio | affects system weight and engine exit pressure (net power), affects heat exchanger size which has only slight effect on incurred drag |
| 2. | Intercooler charge air exit temperature | same effects as above |
| 3. | Inter/aftercooler charge air side pressure drop | trades heat exchanger weight against engine exit pressure (net power), affects heat exchanger size which has only slight effect on incurred drag |
| 4. | Duct mach number | trades duct weight against engine exit pressure (net power), no effect on incurred drag |
| 5. | Power supplied to prop shaft from power turbine | affects power turbine weight (and system weight slightly) and engine exit pressure (net power), no effect on incurred drag |
| 6. | Expansion side duct insulation thickness | trades weight of insulation against engine exit pressure (net power), no effect on incurred drag |
| 7. | Inter/aftercooler air side pressure drop | trades weight of heat exchangers against ram air drag, no effect on engine exit pressure (net power) |
| 8. | Intercooler air flow rate | trades weight of heat exchangers against ram air drag, no effect on engine exit pressure (net power) |

| | | |
|---|---|---|
| 9. | Aftercooler air flow rate | trades weight of heat exchangers against ram air drag, no effect on engine exit pressure (net power) |

Results:

The effects of each of the optimization parameters was examined individually. Several were found to have optima which fell into narrow ranges and could be fixed for system optimization studies.

The dual stage turbocharger system (two compressor stages driven by one turbine stage) was used for evaluating the effects of insulation thickness, pressure ratio split, intercooler charge air exit temperature, and air side pressure drop and flow rate through the intercooler and aftercooler on net drag, net power and equivalent power. Turbocompounding was not included, inter/aftercooler charge air side pressure drop was set to 0.125 psid, and duct Mach number was set to 0.175 for this set of optimization studies.

Insulation Thickness:

As insulation is added to the exhaust ducts, less thermal energy is lost and lower engine back pressure is required to drive the turbines, resulting in a net power increase. However, duct weight and net drag increase. The net effect is for equivalent power to drop as insulation thickness is increased. Therefore, insulation thickness is set to zero since this leads to maximum equivalent power.

Compressor Pressure Ratio Split:

Over a wide range of the other optimization parameters, the first stage compression ratio (Pr) is optimal between 3.6 and 3.8 for maximum equivalent power. Pressure ratio was, therefore, fixed at 3.7.

Ram Air Heat Exchanger Pressure Drop:

As intercooler and aftercooler air side pressure drop ($\Delta$Pc) increases, heat exchanger weight decreases but ram air drag increases. Only net drag is effected. A value of $\Delta$Pc of 0.1 psid was found to be optimal for minimum net drag.

Intercooling:

Figure 11 shows the effect of intercooler charge air exit temperature ($Tic_o$) on net power, net drag and equivalent power. For each value of $Tic_o$, inter/aftercooler ram air flow rate ($Wc_{ic}$ and $Wc_{ac}$) are optimized for minimum net drag (they have no effect on net power). Over a range of $Tic_o$ from 140.0 °F to 210.0 °F the equivalent power is constant within 0.7%. Net drag decreases from 21.71 to 20.11 and net power decreases from 28.19 to 27.74 over this range. $Tic_o$ is set to 150.0 °F since it yields high net power. The values of $Wc_{ic}$ and $Wc_{ac}$ for which net drag is minimum for this temperature are 53.3 and 54.7 lbm/min.

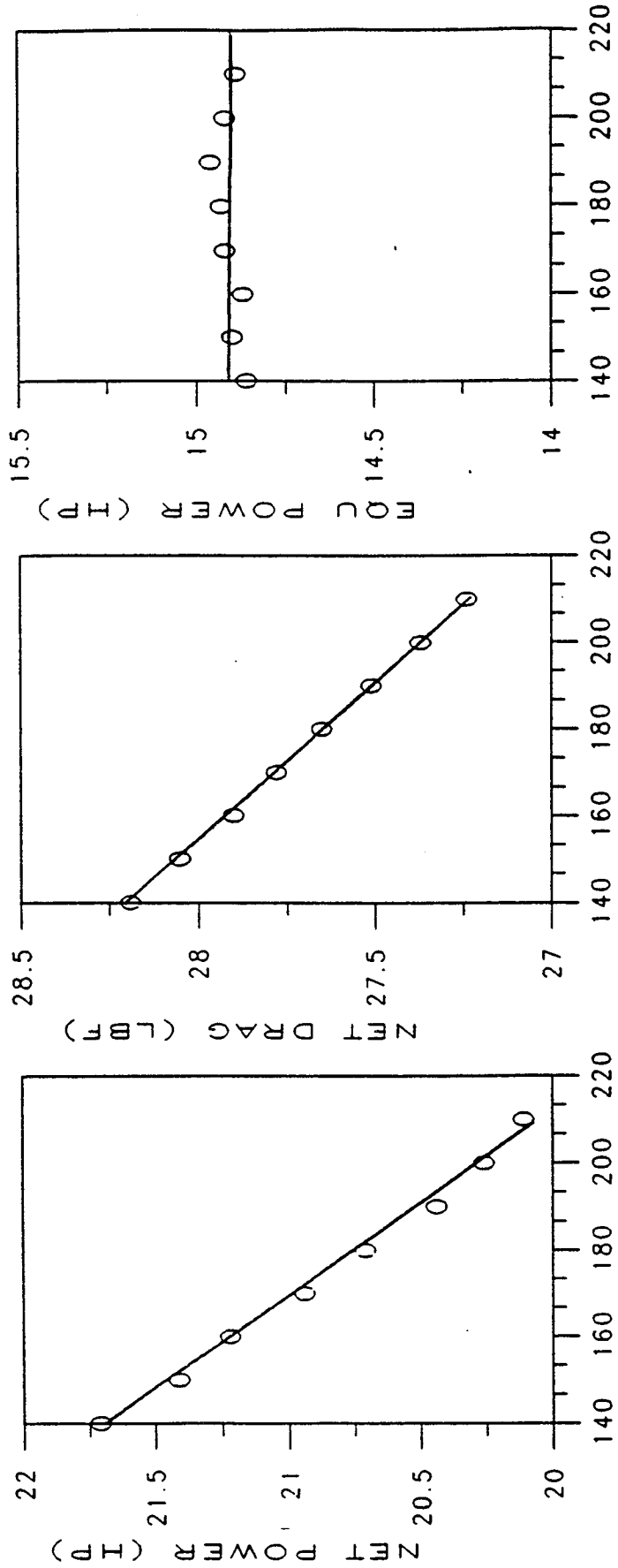



**FIGURE 11. EFFECT OF INTERCOOLING**

Charge Air Heat Exchanger/Duct Pressure Drop:

With the previously discussed optimization parameters fixed, the optimal values of intercooler/aftercooler charge air pressure drop ($\Delta P_{hx}$), duct Mach number ($M_d$), and turbocompounding power remain to be determined. The power turbine is placed upstream of the turbocharger as this results in the lowest weight power turbine, with little effect on the turbocharger. Figure 12 shows the effect of heat exchanger pressure drop and duct Mach number, at various levels of power turbine output power, on net power, net drag, and equivalent power. (The analysis is for steady state flow, analysis with pressure pulse effects is discussed below.) The heat exchanger pressure drop, $\Delta P_{hx}$, and duct Mach number, $M_d$, together characterize the charge air pressure drop. In order to vary them together systematically, normalized parameters are used:

normalized pressure drop, $(\Delta P_{hx} - \Delta P_{hx,min})/(\Delta P_{hx,max} - \Delta P_{hx,min})$

and

normalized duct Mach number, $(M_d - M_{d,min})/(M_{d,max} - M_{d,min})$

The normalized pressure drop and Mach number are increased together along the x-axis of the three plots, from $\Delta P_{hx,min}$ of 0.05 psid to $\Delta P_{hx,max}$ of 0.25 psid, and $M_{d,min}$ of 0.10 to $M_{d,max}$ of 0.30. As the pressure drop and Mach number increase, the heat exchangers and duct weights decrease, which results in a net drag decrease, but power must be expended to overcome the increased charge air pressure drops, and thus net power is decreased. The resultant equivalent power curves are nearly flat with a slight maximum between 25 percent and 50 percent of the pressure drop range, and decrease at higher pressure drops. Values of 0.125 psid and 0.175 Mach number are selected for $\Delta P_{hx}$ and $M_d$.

Power Turbine Power:

Power turbine power is indicated by the coding numbers on the curves, in tens of horsepower (e.g., the curve having the numeral "4" embedded in it indicates a fixed power turbine power of 40 horsepower). With increasing power turbine power, net power increases. Net drag at first increases; there is a jump in weight, and thus in net drag from zero power (no turbine or gearbox) to finite power output. Then net drag decreases as increasing power turbine pressure ratios (at fixed flow) result in faster, smaller turbines. Equivalent power decreases with the initial addition of hardware between zero and non-zero turbine power, then increases. Net and equivalent power increase monotonically with increasing power turbine power, seemingly indicating very large power availability. However, there is probably a practical limit at back pressures sufficiently high (exhaust pressures near engine inlet pressure) to cause engine scavenging difficulties. The inlet and exhaust pressures specified by REDIV (34.9 and 30.6 psia, respectively) are shown on the net power plot. They show a practical power turbine maximum design power to be in the 50-60 horsepower range (with NO pulse effects), with resulting equivalent power of 35-40 horsepower.

Figure 13 shows the effect of $\Delta P_{hx}$ and $M_d$, again using the parameter k, at various levels of turbocompounding power, on net drag, net power, and equivalent power with initial

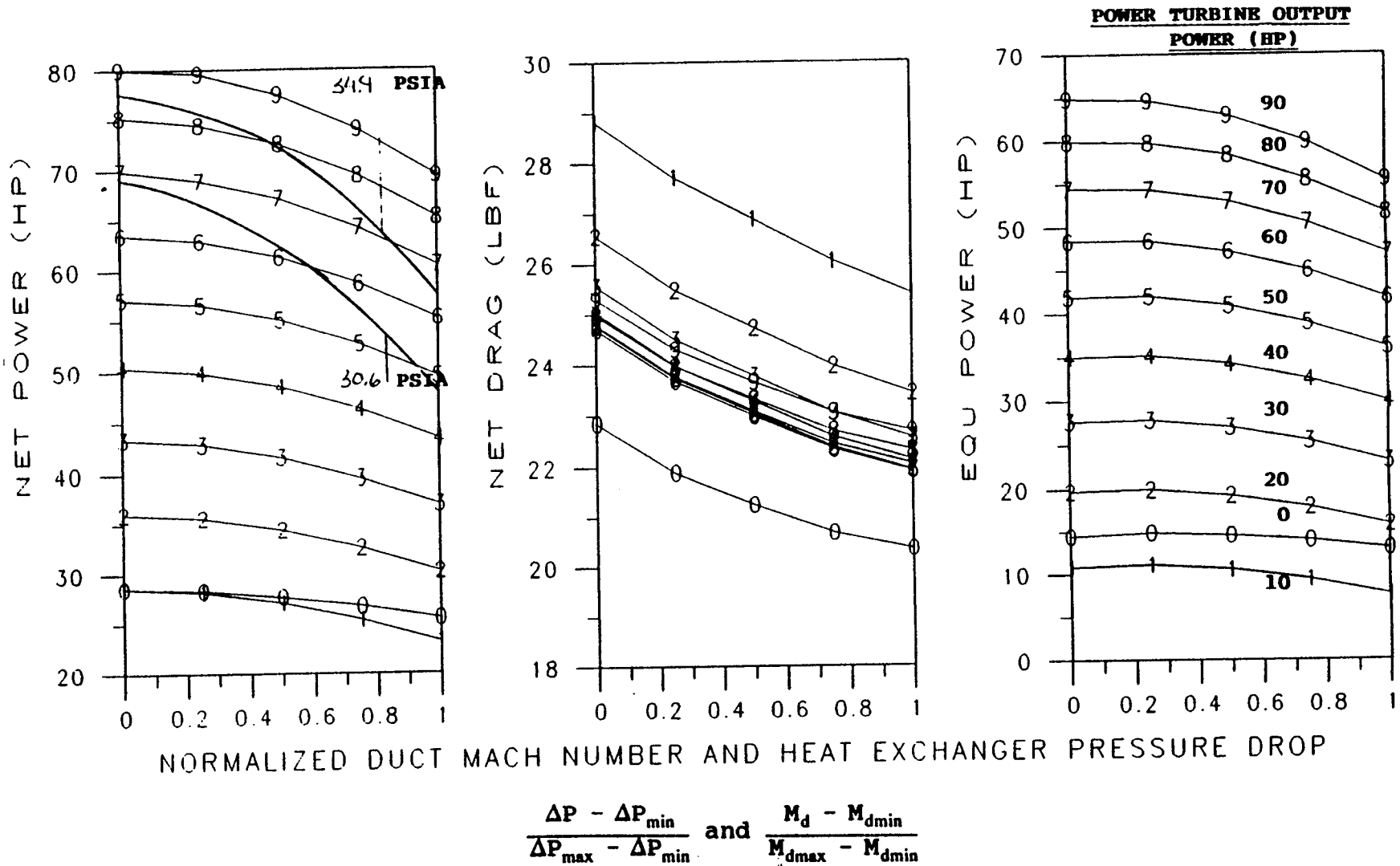



**FIGURE 13. EFFECT OF CHARGE AIR PRESSURE DROP (WITH PULSE EFFECT)**

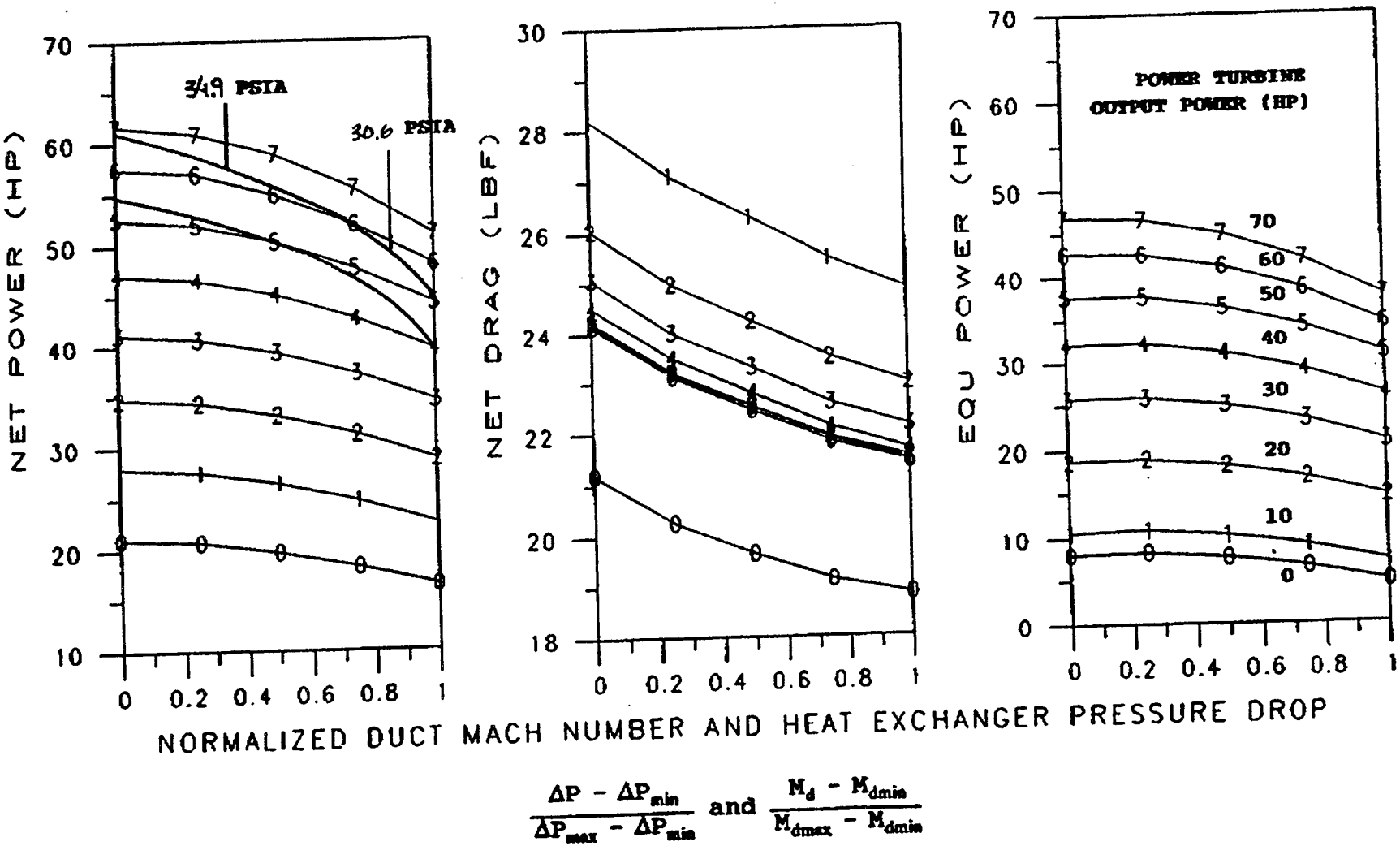



**FIGURE 12. EFFECT OF CHARGE AIR PRESSURE DROP (NO PULSE EFFECTS)**

estimates of pressure pulse effects included. The trends are the same as those with pressure pulse effects neglected, however the levels of available power turbine power are increased to 65-80 horsepower. Both net power and equivalent power are increased by 15-20 horsepower by pulse effects. (Although incomplete, the pressure pulse analysis indicated that even higher turbine powers should be available with pulse recovery.)

In summary, the power turbine analysis shows equivalent power gains of 35 horsepower (minimum, steady flow) to 60 horsepower (maximum, initial pulse analysis), or more. These gains represent recovery of 12-20 percent of the rotary engine shaft power from its wasted exhaust energy; inclusion of the power turbine in the optimum system would seem to be mandatory. Some further pulse analysis, in conjunction with installation design, is needed to quantify the achievable turbocompounding benefit.

## 4.0 RESULTS AND RECOMMENDATIONS

### 4.1 Results

With the optimization parameter ranges characterized, the system analysis program was used to characterize the three candidate systems previously shown in Figure 9. These are the multiple spool, multistage, and single stage turbochargers, each in a system having a power turbine upstream of the turbocharger for turbocompounding. The calculated equivalent powers for the three optimized systems are shown in Figure 14. Power turbine power was fixed at a conservative 40 HP (at the engine output shaft). The dual spool configuration (two turbines driving one compressor each) is thermodynamically equivalent to the dual stage configuration (one turbine driving two compressors) on the compression side so the optimization parameters are optimal at the same values for both. For the single stage configuration (one turbine driving one stage of compression), there is no compressor ratio split to optimize, nor is there an intercooler (only an aftercooler). As in the study with the dual stage turbocharger, the insulation thickness and $\Delta Pc$ optimize to 0.0 and 0.10 respectively, and $\Delta P_{hx}$ and $M_d$ have optimal values of 0.125 psid and 0.175 Mach number. $Wc_{ac}$ is optimized for minimum drag at 99.0 lbm/min.

The dual stage configuration leads to the highest value of equivalent power by a modest amount (16 percent higher than that of the dual spool). The single stage system, expected to be potentially the best, falls midway between the others. Relative to the dual stage configuration, the single stage system suffers from somewhat poorer thermodynamic performance (no intercooling) as well as high aftercooler weight resulting from the need to use steel materials at the high compressor outlet temperature. Data from the turbomachinery system sizing computer program are shown in Tables 3, 4, and 5 for the dual spool, dual stage, and single stage systems, respectively. Figure 15 shows the dual stage configuration system state points.

Since the equivalent power does not indicate a clear advantage for one of the configurations, other factors, such as reliability, cost, off design performance, should be considered. Table 6 lists some of the other advantages and disadvantages of the three configurations.

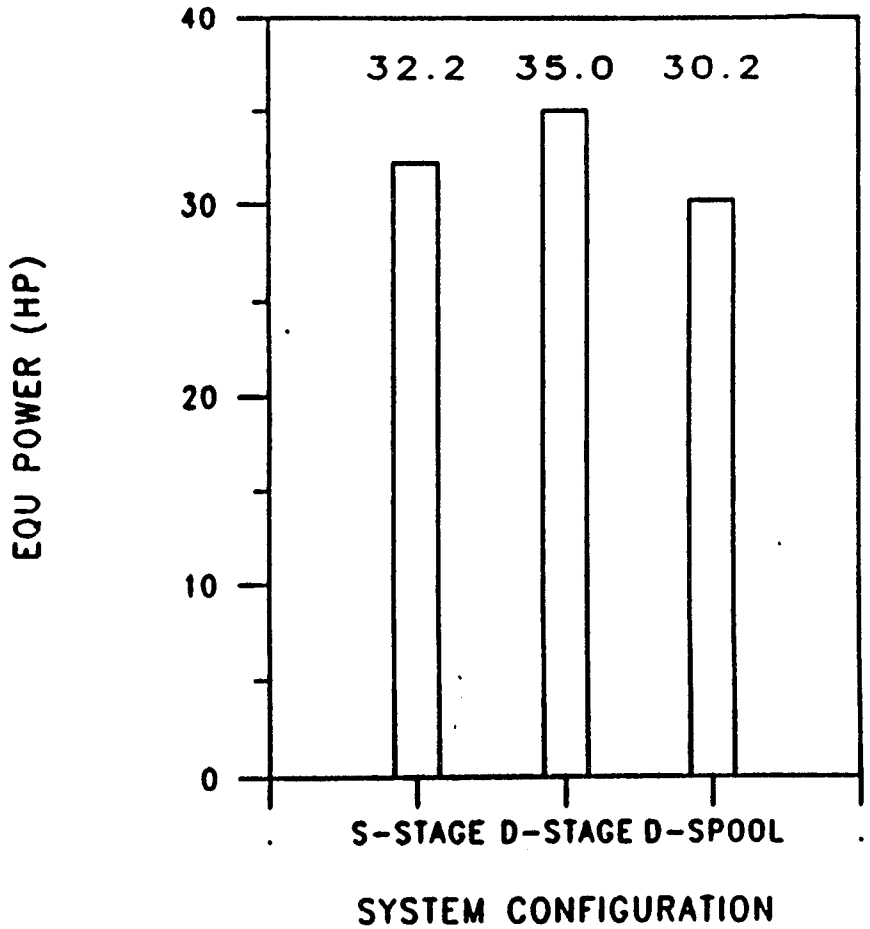



**FIGURE 14. SYSTEM COMPARISON**

**TABLE 3. DUAL SPOOL, 40 HP POWER TURBINE (25 KFT)**

SYSTEM STATE POINTS

| STATION | FLOW LB/MIN | PRESSURE PSIA | TEMP DEG F | MACH | AREA SQ.IN. |
|---|---|---|---|---|---|
| AMBIENT | | 5.46 | 6.71 | 0.2670 | |
| INLET | 70.55 | 5.71 | 12.99 | 0.2029 | 0.1721 |
| COMPRESSOR 1 IN | | 5.70 | 12.99 | 0.1748 | 0.1983 |
| COMPRESSOR 1 OUT | | 21.08 | 282.58 | 0.1750 | 0.0669 |
| INTERCOOLER IN | | 20.97 | 282.58 | 0.1751 | 0.0669 |
| INTERCOOLER OUT | | 20.85 | 150.00 | 0.1750 | 0.0612 |
| COMPRESSOR 2 IN | | 20.74 | 150.00 | 0.1752 | 0.0612 |
| COMPRESSOR 2 OUT | | 35.47 | 278.36 | 0.1750 | 0.0396 |
| AFTERCOOLER IN | | 35.24 | 278.36 | 0.1754 | 0.0396 |
| AFTERCOOLER OUT | | 35.12 | 140.00 | 0.1750 | 0.0360 |
| ENGINE IN | | 34.89 | 140.00 | 0.1755 | 0.0360 |
| ENGINE OUT | | 25.00 | 1154.00 | 0.1750 | 0.0877 |
| TURBINE 4 IN | | 24.92 | 1148.11 | 0.1747 | 0.0877 |
| TURBINE 4 OUT | | 20.10 | 1074.87 | 0.1750 | 0.1062 |
| TURBINE 3 IN | | 19.98 | 1068.86 | 0.1746 | 0.1062 |
| TURBINE 3 OUT | | 13.85 | 951.35 | 0.1750 | 0.1474 |
| TURBINE 2 IN | | 13.78 | 945.17 | 0.1745 | 0.1474 |
| TURBINE 2 OUT | | 5.58 | 690.68 | 0.1750 | 0.0000 |
| TURBINE 1 OUT | | 5.58 | 690.68 | 0.1750 | 0.3286 |
| EXIT | 72.84 | 5.57 | 684.43 | 0.1743 | 0.3286 |
| AMBIENT | | 5.46 | 6.71 | 0.0000 | |

DUCT CONDITIONS AND SIZE

| SECTION | MACH IN | LENGTH FT. | DIA IN. | DP PSI | DT DEG R | WEIGHT LBM | VOLUME CU IN |
|---|---|---|---|---|---|---|---|
| 1 TO 2 | 0.1748 | 3.000 | 6.030 | 0.013 | 0.101 | 2.229 | 1081.5 |
| 3 TO 4 | 0.1750 | 5.000 | 3.501 | 0.110 | 2.446 | 2.106 | 598.6 |
| 5 TO 6 | 0.1750 | 5.000 | 3.350 | 0.110 | 1.253 | 2.020 | 548.8 |
| 7 TO 8 | 0.1750 | 5.000 | 2.695 | 0.228 | 1.905 | 1.631 | 358.4 |
| 9 TO 10 | 0.1750 | 5.000 | 2.571 | 0.228 | 0.920 | 1.561 | 326.7 |
| 11 TO 12 | 0.1750 | 3.000 | 4.010 | 0.074 | 5.888 | 4.139 | 468.9 |
| 13 TO 14 | 0.1750 | 3.000 | 4.413 | 0.055 | 6.007 | 4.552 | 566.3 |
| 15 TO 16 | 0.1750 | 3.000 | 5.199 | 0.033 | 6.183 | 5.355 | 782.7 |
| 19 TO 20 | 0.1750 | 3.000 | 7.762 | 0.010 | 6.250 | 7.974 | 1730.8 |

HEAT EXCHANGER CONDITIONS AND SIZE

INTERCOOLER

| | SINK | CHARGE AIR |
|---|---|---|
| INLET TEMPERATURE (F) | 15.23 | 282.58 |
| OUTLET TEMPERATURE (F) | 190.40 | 150.00 |
| PRESSURE DROP SPECIFIED (PSIA) | 0.10 | 0.13 |
| PRESSURE DROP ACTUAL (PSIA) | 0.10 | 0.12 |
| FLOW RATE (LBM/MIN) | 53.30 | 70.55 |
| EFFECTIVENESS / LOAD (BTU/MIN) | 0.4959 | 2239. |
| WEIGHT (LBM) / VOLUME (IN3) | 19.93 | 770.2 |

**TABLE 3.** DUAL SPOOL (continued)

AFTERCOOLER

| | SINK | CHARGE AIR |
|---|---|---|
| INLET TEMPERATURE (F) | 15.23 | 278.36 |
| OUTLET TEMPERATURE (F) | 193.36 | 140.00 |
| PRESSURE DROP SPECIFIED (PSIA) | 0.10 | 0.13 |
| PRESSURE DROP ACTUAL (PSIA) | 0.10 | 0.13 |
| FLOW RATE (LBM/MIN) | 54.70 | 70.55 |
| EFFECTIVENESS / LOAD (BTU/MIN) | 0.5258 | 2336. |
| WEIGHT (LBM) / VOLUME (IN3) | 22.46 | 900.4 |

COMPRESSOR/TURBINE CONDITIONS AND SIZE

COMPRESSION STAGES = 2

| STAGE | 1 | 2 | 3 | 4 |
|---|---|---|---|---|
| SHAFT POWER, BTU/MIN | 0.00 | 4556.22 | 2154.31 | 1696.67 |
| COMPRESSOR CP, BTU/LBM F | 0.0000 | 0.2410 | 0.2394 | 0.0000 |
| COMP DELTA H, BTU/MIN | 0.00 | 64.58 | 30.54 | 0.00 |

EXPANSION STAGES = 3

| STAGE | 1 | 2 | 3 | 4 |
|---|---|---|---|---|
| TURBINE CP, BTU/LBM F | 0.0000 | 0.2602 | 0.2663 | 0.2692 |
| TURB DELTA H, BTU/MIN | 0.00 | 65.84 | 31.13 | 24.52 |
| AVG PRESS DH, BTU/MIN | 0.00 | 65.84 | 31.13 | 19.62 |

POWER TURBINE SHAFT 4

| | TURBINE |
|---|---|
| PRESSURE RATIO | 1.24 |
| SPEED (RPM) | 23295. |
| SPECIFIC SPEED | 0.6000 |
| DIAMETER (IN) | 7.879 |
| WEIGHT (LBM) | 34.89 |
| VOLUME (IN3) | 2569. |

TURBOCHARGER SHAFT 2 WITH SINGLE STAGE COMPRESSOR

| | TURBINE | COMPRESSOR |
|---|---|---|
| PRESSURE RATIO | 2.47 | 3.70 |
| SPEED (RPM) | 30918. | |
| SPECIFIC SPEED | 0.6000 | 0.5099 |
| DIAMETER (IN) | 9.728 | 11.430 |
| WEIGHT (LBM) | 53.18 | 52.26 |
| VOLUME (IN3) | 5969. | 11380. |

TURBOCHARGER SHAFT 3 WITH SINGLE STAGE COMPRESSOR

| | TURBINE | COMPRESSOR |
|---|---|---|
| PRESSURE RATIO | 1.44 | 1.71 |
| SPEED (RPM) | 29041. | |
| SPECIFIC SPEED | 0.6947 | 0.5000 |
| DIAMETER (IN) | 7.342 | 8.368 |
| WEIGHT (LBM) | 30.29 | 28.01 |
| VOLUME (IN3) | 1937. | 3268. |

**TABLE 3.** DUAL SPOOL (continued)

WASTE GATE SIZING

| | | |
|---|---|---|
| WASTE GATE 3 | CORRECTED FLOW | 0.9860 |
| | WEIGHT (LBM) | 6.06 |
| WASTE GATE 2 | CORRECTED FLOW | 1.3717 |
| | WEIGHT (LBM) | 9.95 |

GEAR BOX SIZING

| | |
|---|---|
| PROP POWER | 40.00 |
| WEIGHT (LBM) | 9.880 |

SYSTEM SUMMARY

TURBOMACHINERY SYSTEM NET DELIVERED POWER

| PEOUT | ENG DELTA | POWER TRB | NET |
|---|---|---|---|
| 25.00 | 9.62 | 40.00 | 49.62 |

SYSTEM WEIGHT, DRAG AND NET DRAG

| WEIGHT | DRAG | NET DRAG | VAMB |
|---|---|---|---|
| 298.47 | 5.05 | 31.70 | 280.38 |

NET EQUIVALENT POWER = 30.16

**TABLE 4.** DUAL STAGE, 40 HP POWER TURBINE (25 KFT)

SYSTEM STATE POINTS

| STATION | FLOW LB/MIN | PRESSURE PSIA | TEMP DEG F | MACH | AREA SQ.IN. |
|---|---|---|---|---|---|
| AMBIENT | | 5.46 | 6.71 | 0.2670 | |
| INLET | 70.55 | 5.71 | 12.99 | 0.2029 | 0.1721 |
| COMPRESSOR 1 IN | | 5.70 | 12.99 | 0.1748 | 0.1983 |
| COMPRESSOR 1 OUT | | 21.08 | 282.58 | 0.1750 | 0.0669 |
| INTERCOOLER IN | | 20.97 | 282.58 | 0.1751 | 0.0669 |
| INTERCOOLER OUT | | 20.85 | 150.00 | 0.1750 | 0.0612 |
| COMPRESSOR 2 IN | | 20.74 | 150.00 | 0.1752 | 0.0612 |
| COMPRESSOR 2 OUT | | 35.47 | 278.36 | 0.1750 | 0.0396 |
| AFTERCOOLER IN | | 35.24 | 278.36 | 0.1754 | 0.0396 |
| AFTERCOOLER OUT | | 35.12 | 140.00 | 0.1750 | 0.0360 |
| ENGINE IN | | 34.89 | 140.00 | 0.1755 | 0.0360 |
| ENGINE OUT | | 25.14 | 1154.00 | 0.1750 | 0.0872 |
| TURBINE 4 IN | | 25.07 | 1148.13 | 0.1747 | 0.0872 |
| TURBINE 4 OUT | | 20.22 | 1074.89 | 0.1750 | 0.0000 |
| TURBINE 3 OUT | | 20.22 | 1074.89 | 0.1750 | 0.1056 |
| TURBINE 2 IN | | 20.10 | 1068.89 | 0.1746 | 0.1056 |
| TURBINE 2 OUT | | 5.58 | 697.11 | 0.1750 | 0.0000 |
| TURBINE 1 OUT | | 5.58 | 697.11 | 0.1750 | 0.3295 |
| EXIT | 72.84 | 5.57 | 690.80 | 0.1743 | 0.3295 |
| AMBIENT | | 5.46 | 6.71 | 0.0000 | |

DUCT CONDITIONS AND SIZE

| SECTION | MACH IN | LENGTH FT. | DIA IN. | DP PSI | DT DEG R | WEIGHT LBM | VOLUME CU IN |
|---|---|---|---|---|---|---|---|
| 1 TO 2 | 0.1748 | 3.000 | 6.030 | 0.013 | 0.101 | 2.229 | 1081.5 |
| 3 TO 4 | 0.1750 | 5.000 | 3.501 | 0.110 | 2.446 | 2.106 | 598.6 |
| 5 TO 6 | 0.1750 | 5.000 | 3.350 | 0.110 | 1.253 | 2.020 | 548.8 |
| 7 TO 8 | 0.1750 | 5.000 | 2.695 | 0.228 | 1.905 | 1.631 | 358.4 |
| 9 TO 10 | 0.1750 | 5.000 | 2.571 | 0.228 | 0.920 | 1.561 | 326.7 |
| 11 TO 12 | 0.1750 | 3.000 | 3.998 | 0.075 | 5.873 | 4.127 | 466.1 |
| 15 TO 16 | 0.1750 | 3.000 | 4.400 | 0.055 | 5.992 | 4.538 | 563.1 |
| 19 TO 20 | 0.1750 | 3.000 | 7.772 | 0.010 | 6.309 | 7.985 | 1735.5 |

HEAT EXCHANGER CONDITIONS AND SIZE

INTERCOOLER

| | SINK | CHARGE AIR |
|---|---|---|
| INLET TEMPERATURE (F) | 15.23 | 282.58 |
| OUTLET TEMPERATURE (F) | 190.40 | 150.00 |
| PRESSURE DROP SPECIFIED (PSIA) | 0.10 | 0.13 |
| PRESSURE DROP ACTUAL (PSIA) | 0.10 | 0.12 |
| FLOW RATE (LBM/MIN) | 53.30 | 70.55 |
| EFFECTIVENESS / LOAD (BTU/MIN) | 0.4959 | 2239. |
| WEIGHT (LBM) / VOLUME (IN3) | 19.93 | 770.2 |

**TABLE 4.** DUAL STAGE (continued)

AFTERCOOLER

| | SINK | CHARGE AIR |
|---|---|---|
| INLET TEMPERATURE (F) | 15.23 | 278.36 |
| OUTLET TEMPERATURE (F) | 193.36 | 140.00 |
| PRESSURE DROP SPECIFIED (PSIA) | 0.10 | 0.13 |
| PRESSURE DROP ACTUAL (PSIA) | 0.10 | 0.13 |
| FLOW RATE (LBM/MIN) | 54.70 | 70.55 |
| EFFECTIVENESS / LOAD (BTU/MIN) | 0.5258 | 2336. |
| WEIGHT (LBM) / VOLUME (IN3) | 22.46 | 900.4 |

COMPRESSOR/TURBINE CONDITIONS AND SIZE

COMPRESSION STAGES = 2

| STAGE | 1 | 2 | 3 | 4 |
|---|---|---|---|---|
| SHAFT POWER, BTU/MIN | 0.00 | 6710.53 | 0.00 | 1696.67 |
| COMPRESSOR CP, BTU/LBM F | 0.0000 | 0.2410 | 0.2394 | 0.0000 |
| COMPR DELTA H, BTU/MIN | 0.00 | 64.58 | 30.54 | 0.00 |

EXPANSION STAGES = 2

| STAGE | 1 | 2 | 3 | 4 |
|---|---|---|---|---|
| TURBINE CP, BTU/LBM F | 0.0000 | 0.2623 | 0.0000 | 0.2692 |
| TURB DELTA H, BTU/MIN | 0.00 | 96.98 | 0.00 | 24.52 |
| AVG PRESS DH, BTU/MIN | 0.00 | 96.98 | 0.00 | 19.62 |

POWER TURBINE SHAFT 4

| | TURBINE |
|---|---|
| PRESSURE RATIO | 1.24 |
| SPEED (RPM) | 23363. |
| SPECIFIC SPEED | 0.6000 |
| DIAMETER (IN) | 7.856 |
| WEIGHT (LBM) | 34.68 |
| VOLUME (IN3) | 2539. |

TURBOCHARGER SHAFT 2 WITH DUAL STAGE COMPRESSOR

| | TURBINE | COMP 1 | COMP 2 |
|---|---|---|---|
| PRESSURE RATIO | 3.60 | 3.70 | 1.71 |
| SPEED (RPM) | 41226. | | |
| SPECIFIC SPEED | 0.6000 | 0.6798 | 0.7098 |
| DIAMETER (IN) | 8.854 | 8.572 | 5.895 |
| WEIGHT (LBM) | 44.05 | | 43.29 |
| VOLUME (IN3) | 4097. | 3600. | 805. |

WASTE GATE SIZING

| | | |
|---|---|---|
| WASTE GATE 2 | CORRECTED FLOW | 0.9802 |
| | WEIGHT (LBM) | 6.01 |

GEAR BOX SIZING

| | |
|---|---|
| PROP POWER | 40.00 |
| WEIGHT (LBM) | 9.880 |

**TABLE 4.** DUAL STAGE (continued)

SYSTEM SUMMARY

TURBOMACHINERY SYSTEM NET DELIVERED POWER

| PEOUT | ENG DELTA | POWER TRB | NET |
|---|---|---|---|
| 25.14 | 9.37 | 40.00 | 49.37 |

SYSTEM WEIGHT, DRAG AND NET DRAG

| WEIGHT | DRAG | NET DRAG | VAMB |
|---|---|---|---|
| 206.50 | 5.05 | 23.48 | 280.38 |

NET EQUIVALENT POWER = 34.95

**TABLE 5.** SINGLE STAGE, 40 HP POWER TURBINE (25 KFT)

SYSTEM STATE POINTS

| STATION | FLOW LB/MIN | PRESSURE PSIA | TEMP DEG F | MACH | AREA SQ.IN. |
|---|---|---|---|---|---|
| AMBIENT | | 5.46 | 6.71 | 0.2670 | |
| INLET | 70.55 | 5.71 | 12.99 | 0.2029 | 0.1721 |
| COMPRESSOR 1 IN | | 5.70 | 12.99 | 0.1748 | 0.1983 |
| COMPRESSOR 1 OUT | | 35.47 | 427.71 | 0.1750 | 0.0436 |
| INTERCOOLER IN | | 35.24 | 427.71 | 0.1753 | 0.0436 |
| INTERCOOLER OUT | | 35.12 | 140.00 | 0.1750 | 0.0360 |
| ENGINE IN | | 34.89 | 140.00 | 0.1755 | 0.0360 |
| ENGINE OUT | | 27.04 | 1154.00 | 0.1750 | 0.0811 |
| TURBINE 4 IN | | 26.96 | 1148.31 | 0.1747 | 0.0811 |
| TURBINE 4 OUT | | 21.74 | 1075.07 | 0.1750 | 0.0000 |
| TURBINE 3 OUT | | 21.74 | 1075.07 | 0.1750 | 0.0982 |
| TURBINE 2 IN | | 21.61 | 1069.27 | 0.1746 | 0.0982 |
| TURBINE 2 OUT | | 5.58 | 679.47 | 0.1750 | 0.0000 |
| TURBINE 1 OUT | | 5.58 | 679.47 | 0.1750 | 0.3268 |
| EXIT | 72.84 | 5.57 | 673.32 | 0.1743 | 0.3268 |
| AMBIENT | | 5.46 | 6.71 | 0.0000 | |

DUCT CONDITIONS AND SIZE

| SECTION | MACH IN | LENGTH FT. | DIA IN. | DP PSI | DT DEG R | WEIGHT LBM | VOLUME CU IN |
|---|---|---|---|---|---|---|---|
| 1 TO 2 | 0.1748 | 3.000 | 6.030 | 0.013 | 0.101 | 2.229 | 1081.5 |
| 3 TO 4 | 0.1750 | 5.000 | 2.828 | 0.223 | 3.001 | 1.708 | 393.8 |
| 5 TO 6 | 0.1750 | 5.000 | 2.571 | 0.228 | 0.920 | 1.561 | 326.7 |
| 11 TO 12 | 0.1750 | 3.000 | 3.855 | 0.083 | 5.686 | 3.981 | 433.9 |
| 15 TO 16 | 0.1750 | 3.000 | 4.243 | 0.061 | 5.807 | 4.378 | 524.1 |
| 19 TO 20 | 0.1750 | 3.000 | 7.740 | 0.010 | 6.146 | 7.952 | 1721.4 |

HEAT EXCHANGER CONDITIONS AND SIZE

HIGH TEMP AFTERCOOLER

| | SINK | CHARGE AIR |
|---|---|---|
| INLET TEMPERATURE (F) | 15.23 | 427.71 |
| OUTLET TEMPERATURE (F) | 221.72 | 349.99 |
| PRESSURE DROP SPECIFIED (PSIA) | 0.10 | 0.03 |
| PRESSURE DROP ACTUAL (PSIA) | 0.10 | 0.03 |
| FLOW RATE (LBM/MIN) | 26.74 | 70.55 |
| EFFECTIVENESS / LOAD (BTU/MIN) | 0.1884 | 1322. |
| WEIGHT (LBM) / VOLUME (IN3) | 20.82 | 416.1 |

**TABLE 5.** SINGLE STAGE (continued)

LOW TEMP AFTERCOOLER

| | SINK | CHARGE AIR |
|---|---|---|
| INLET TEMPERATURE (F) | 15.23 | 349.99 |
| OUTLET TEMPERATURE (F) | 221.72 | 140.00 |
| PRESSURE DROP SPECIFIED (PSIA) | 0.10 | 0.09 |
| PRESSURE DROP ACTUAL (PSIA) | 0.10 | 0.09 |
| FLOW RATE (LBM/MIN) | 72.26 | 70.55 |
| EFFECTIVENESS / LOAD (BTU/MIN) | 0.6273 | 3572. |
| WEIGHT (LBM) / VOLUME (IN3) | 30.55 | 1233.4 |

COMPRESSOR/TURBINE CONDITIONS AND SIZE

COMPRESSION STAGES = 1

| STAGE | 1 | 2 | 3 | 4 |
|---|---|---|---|---|
| SHAFT POWER, BTU/MIN | 0.00 | 7027.64 | 0.00 | 1696.67 |
| COMPRESSOR CP, BTU/LBM F | 0.0000 | 0.2417 | 0.0000 | 0.0000 |
| COMPR DELTA H, BTU/MIN | 0.00 | 99.61 | 0.00 | 0.00 |

EXPANSION STAGES = 2

| STAGE | 1 | 2 | 3 | 4 |
|---|---|---|---|---|
| TURBINE CP, BTU/LBM F | 0.0000 | 0.2620 | 0.0000 | 0.2692 |
| TURB DELTA H, BTU/MIN | 0.00 | 101.56 | 0.00 | 24.52 |
| AVG PRESS DH, BTU/MIN | 0.00 | 101.56 | 0.00 | 19.62 |

POWER TURBINE SHAFT 4

| | TURBINE |
|---|---|
| PRESSURE RATIO | 1.24 |
| SPEED (RPM) | 24227. |
| SPECIFIC SPEED | 0.6000 |
| DIAMETER (IN) | 7.576 |
| WEIGHT (LBM) | 32.25 |
| VOLUME (IN3) | 2196. |

TURBOCHARGER SHAFT 2 WITH SINGLE STAGE COMPRESSOR

| | TURBINE | COMPRESSOR |
|---|---|---|
| PRESSURE RATIO | 3.87 | 6.22 |
| SPEED (RPM) | 43010. | |
| SPECIFIC SPEED | 0.6000 | 0.5198 |
| DIAMETER (IN) | 8.685 | 10.205 |
| WEIGHT (LBM) | 42.39 | 41.65 |
| VOLUME (IN3) | 3792. | 7230. |

WASTE GATE SIZING

| | | |
|---|---|---|
| WASTE GATE 2 | CORRECTED FLOW | 0.9115 |
| | WEIGHT (LBM) | 5.39 |

GEAR BOX SIZING

| | |
|---|---|
| PROP POWER | 40.00 |
| WEIGHT (LBM) | 9.88 |

**TABLE 5.** SINGLE STAGE (continued)

SYSTEM SUMMARY

TURBOMACHINERY SYSTEM NET DELIVERED POWER

| PEOUT | ENG DELTA | POWER TRB | NET |
|---|---|---|---|
| 27.04 | 6.11 | 40.00 | 46.11 |

SYSTEM WEIGHT, DRAG AND NET DRAG

| WEIGHT | DRAG | NET DRAG | VAMB |
|---|---|---|---|
| 204.74 | 4.37 | 22.65 | 280.38 |

NET EQUIVALENT POWER = 32.20



C-3

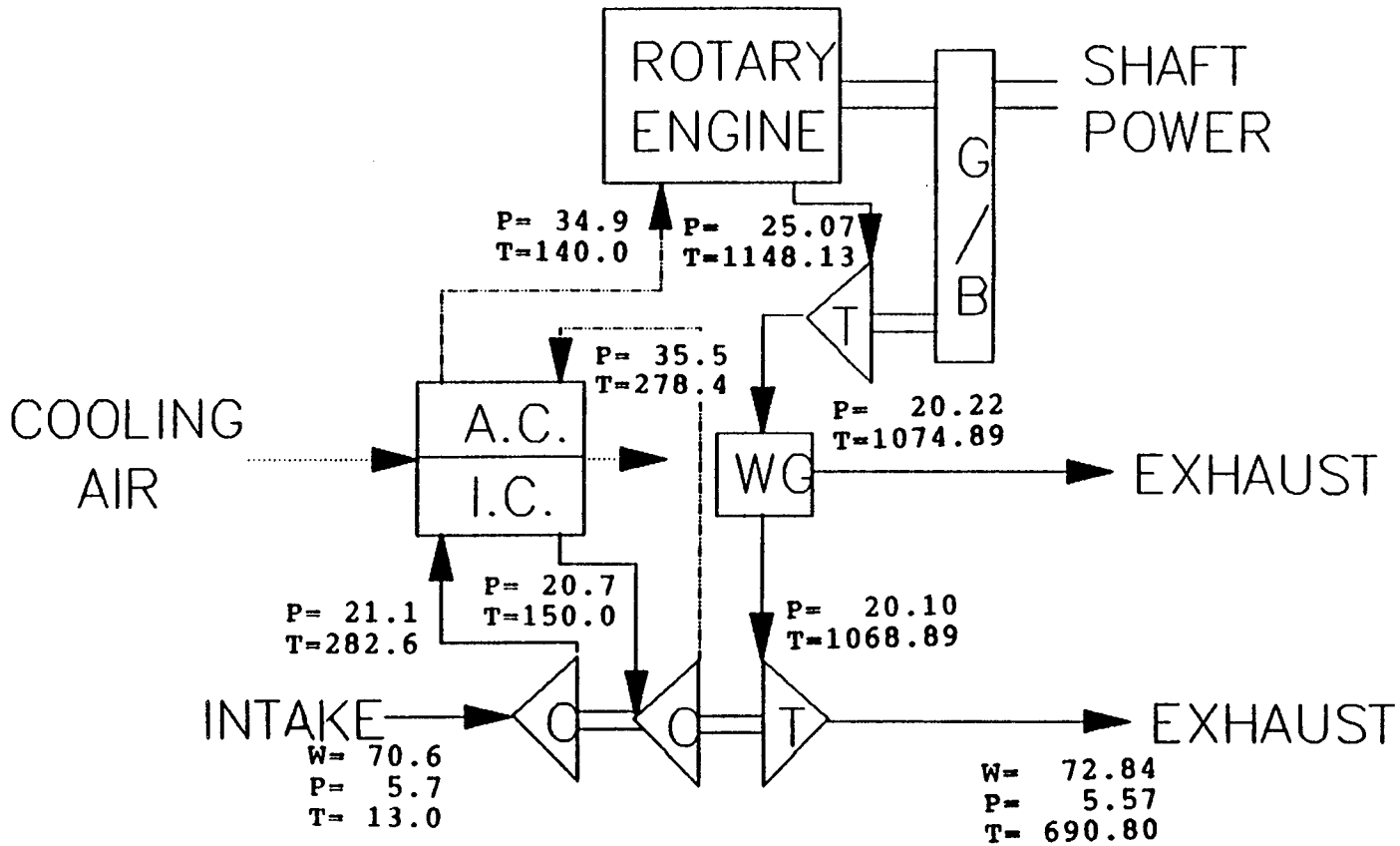



**FIGURE 15. SYSTEM STATE POINTS**

Table 6

| | |
|---|---|
| Dual spool | - largest region of shaft speed where speed match is possible. Allows each turbine and compressor to be run at near optimal speed for best efficiency<br>- lowest engine back pressure<br>- good performance range for off-design operation<br>- most complex, highest weight |
| Dual stage | - smaller region of shaft speed where speed match is possible. More difficult to get all turbomachinery elements to optimal speed for best efficiency<br>- good performance range for off-design operation<br>- low engine back pressure<br>- simple and low weight |
| Single stage | - simple and low weight<br>- high engine back pressure<br>- narrower performance range for off-design operation<br>- very narrow region of shaft speed where speed match is possible. May not be possible to get both wheels to optimal speed for best efficiency<br>- high compressor and turbine tip speeds (high stress) |

## 4.2 Recommendations

The dual stage turbocharging configuration with an upstream turbocompounding (power) turbine is preferred for its high equivalent power, good off-design operating performance range, and reasonable simplicity. The recovery of pulse power leads to substantial increases in power turbine output, so the system should be designed to fully utilize this power. The power turbine can provide at least 60 to 80 hp, while keeping the engine back pressure between 30.6 psia and 34.9 psia.

Additional analytical efforts are needed to better quantify exhaust pulse effects, which apparently will significantly affect turbomachinery sizing and power output. Additionally, an off-design system analysis should be performed to learn how the proposed system configuration(s) perform at lower altitudes and lower throttle settings. These conditions will determine control schedules and may have an impact on proper turbomachinery matching.

## 5.0 REFERENCES

1. Kays, William M. and London, A. L., Compact Heat Exchangers, Third Edition, (McGraw-Hill Book Company, 1984).

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C.E. Irion and R.E. Mount

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**13. ABSTRACT** ***(Maximum 200 words)***

This report summarizes results of a critical technology enablement effort with the stratified charge rotary engine (SCRE) focusing on a power section of 0.67 liters (40 cu. in.) per rotor in single and two rotor versions. The work is a continuation of prior NASA Contracts NAS3–23056 and NAS3–24628. Technical objectives are multi-fuel capability, including civil and military jet fuel and DF-2, fuel efficiency of 0.355 Lbs/BHP-Hr. at best cruise condition above 50% power, altitude capability of up to 10Km (33,000 ft.) cruise, 2000 hour TBO and reduced coolant heat rejection. Critical technologies for SCREs that have the potential for competitive performance and cost in a representative light-aircraft environment were examined. Objectives were: the development and utilization of advanced analytical tools, i.e. higher speed and enhanced three dimensional combustion modeling; identification of critical technologies; development of improved instrumentation, and to isolate and quantitatively identify the contribution to performance and efficiency of critical components or subsystems.

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