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"LOW COST" MICROTURBINES VIA THE TURBOCHARGER ROUTE.

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ABSTRACT

Hosts of small start up companies and private institutions have become and are being enamored with the apparent feasibility of readily converting small turbochargers (T/C's) into low cost microturbines. Turbocharger to micro turbine convertibility has indeed been demonstrated primarily motivated as an opening bid to gain recognized credentials into the potentially huge small turbogenerator commercial market.

It is coincidently opportune that advanced technology development of the small T/C with a single stage centrifugal compressor and single stage radial inflow turbine is currently under going a meta-morphis in order to satisfy green environment mandated automobile fuel economy and emissions goals.

The similarity of the T/C to the conventionally adopted microturbine configuration with radial compressor and turbine components plus undergoing T/C technology advancement entices the virtues of low manufacturing cost attributes associated with the millions of T/Cs that are produced annually.

This paper examines the viability of these converted T/C efforts in attempting to penetrate the small gas turbine market, versus those more specially developed, albeit more costly units, already being marketed by established microturbine manufacturers.

INTRODUCTION

During the late 1950's it became popular to add turbochargers (T/C) to truck diesels as a means of power boosting, followed in the 1960's by application to small automobiles as a means of reducing engine weight and under-hood volume. As a consequence centrifugal compressor research experienced an impetus to increase performance, together with improved manufacturing techniques and affordability. A new phase of T/C turbomachinery technology now evolves in order to meet government mandated automobile fuel economy and emissions goal through which improvements in both compressor efficiency and surge margin are being realized and extended to diminutive T/Cs with rotational speeds approaching 300 krpm. The overwhelming motivations to comply with this mandate are however power boosting to reduce engine weight, thus vehicle tractive effort, and exhaust gas recirculation.

T/C manufacturing cost constraints still favor the frame size or family concept of T/Cs in which several basic impeller sizes are produced with trim variations within each frame size to match a variety of individual customer applications.

Cost typically mandates the baseline frame design as a moderately high specific speed (Ns) compressor and turbine design as Fig 1in order to minimize size (thus weight and cost) for a given airflow.

NOMENCLATURE

D	Diameter
ETATH	Thermal Efficiency
EGT	Exhaust Gas Temperature
g	Gravity
Had	Adiabatic Head
CHP	Combined Heat and Power
Kw	Kilowatt electrical
Ns	Specific Speed
	$= \omega(W / \rho)^{0.5} / (g\ Had)^{0.75}$
R	Pressure Ratio
SFC	Specific Fuel consumption
T/C	Turbocharger
TIT	Turbine Inlet Temperature
Vo	Theoretical Spouting $(2g\ Had)^{0.5}$
W	Airflow
ρ	Density
µ	Kinematic Viscosity
ω	Angular Velocity
ε	Recuperator Effectiveness
Subscripts	
ad	Adiabatic

c Compressor
t Turbine
m Mechanical

Keywords: Microturbine. Turbocharger, Low Emissions.

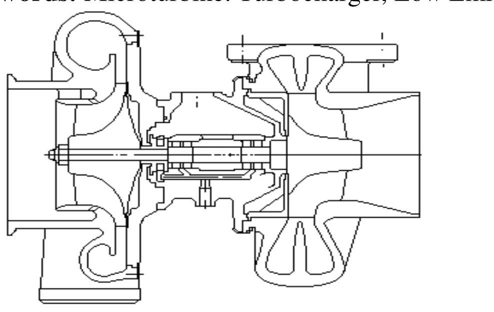

Fig 1. Typical Small Turbocharger.

Although the rotating assembly of this T/C, at first glance, has much in common with the single shaft radial flow small recuperated microturbine the initial design requirements are so significantly different, as to impair the basic concept of convertibility. The intent of this paper is to reveal how these design differences may impact the viability of a small T/C being considered for conversion into a "low cost" microturbine" to serve two prominent markets small generator sets and potentially hybrid car battery chargers.

This conversion has been demonstrated in many instances, Visser.W [1], even by homebuilt enthusiasts, the most popular two of which are portrayed schematically on Fig 2 both with a single can combustor and one incorporating an "add on" box type recuperator .

Note for illustration purposes the combustor and recuperator are depicted significantly reduced in size.

2. TURBOCHARGER DESIGN CONSIDERATIONS.

Some of the design constraints differences between T/C centrifugal compressors and radial turbines as compared to microturbines are compiled on Table 1and 2. Both T/Cs and microturbines should possess adequate compressor surge margin for acceleration and rapid load changes yet microturbine flow range requirements are inherently lower and normally capable of handling such transients without inducer bleed recirculation and or variable compressor geometry both of which may incur compressor efficiency penalties. Most small T/Cs for vehicular applications utilize vaneless diffusers with a scroll casing to maximize flow range and reduce cost, whereas larger T/Cs for mainly diesel generator sets are now beginning to incorporate improved flow range vaned diffusers.

The major performances parameters influencing the performance of reciprocating engine small turbochargers are basically:

- Component Efficiencies
- Engine cylinder pressure drop
- Pressure Ratio and airflow.

The variation of these parameters is dependent upon engine load and speed in addition to which adequate compressor surge margin and overboost protection are required. The turbine volute tongue or nozzle area, and component efficiencies essentially govern the engine back pressure and EGT. The component efficiencies are related to the rotor sizes, specific speeds, and clearance gaps, and surface finishes, plus viscous drags in the bearings and seals. A simple steady state T/C compressor and turbine work balance reveals that EGT is a function of the component efficiencies, massflow and pressure drop ratios, i.e.,

$$EGT\ f\,(\eta c\ \eta t\ \ \eta m)*(Rc/Rt)*(Wt/Wc) \qquad \text{-------- (1)}$$

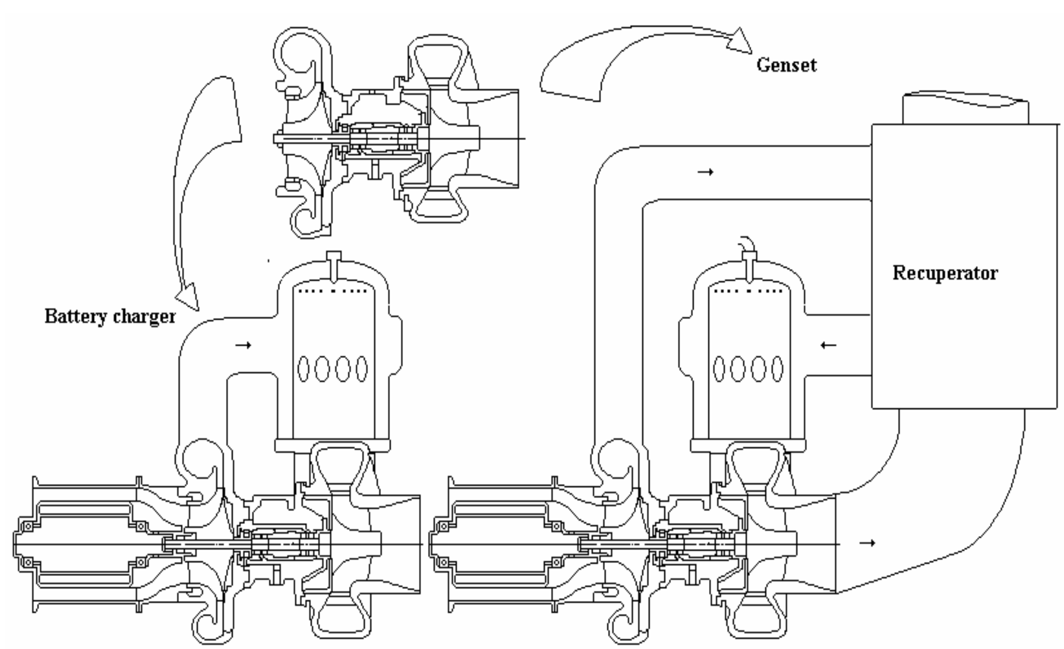

Fig 2. Turbocharger to Microturbine.

Table 1.	**Centrifugal Compressor Design Constraints.**	
Item	**T/C**	Small Gas Turbine
Material	Alum Alloy	Alum/Titanium/Steel
Geometry	Limited by casting method	Limited by Stress and 5 axis m/c
Blade number	Typically 12- 16	Typically 16-30.
Inducer Flap	> 4 to 5 x 1st order Hz	Min 2 x 1st order Hz
Hub diameter	Min as limited by torque nut	Larger for front drive power output
Flow Range	Normally mandates vaneless diffuser and scroll	High vane number diffuser
Pressure Ratio	Max 4.0 (short time)	Up to 8.0 :1
Tip clearance	0.4% tip blade height	0.2% tip blade height
Efficiency	Surge margin dependent	Max possible
Specific Speed Ns	0.9-1.1	0.7-0.9

Table 2.	**Radial Turbine Design Constraints**	
Item	T/C	Small Gas Turbine
TIT	Diesel 1500 F Spark Ignition 1700 F	1700-1900 F
Material.	GMR235-In 713.	Udimet 720-Mar M 247.
Geometry	Limited by radial pull casting	Individual blade waxes permissible
Blade Number	Typically 10-12	Typically 12-16
Exducer Flap	> 4 to 5 x 1st order Hz	Min 2 * 1st order Hz
Tip Speed U	Max 1700 fps	Max 2100 fps
Velocity ratio U/V	0.55 -0.65	0.65 – 0.75
Nozzle	Volute type	Vaned type
Exhaust Diffuser	Minimal	Large area ratio
Tip clearance	0.5% tip blade height	0.3% tip blade height
Efficiency	Typically 70-76%	Peak 86 %

This relationship together with centrifugal compressor and radial inflow turbine specific speed characteristics, Rodgers.C., Rochford.K., [2] can be used for preliminary optimization of turbocharger relative frame size over a desired flow range. The optimizing parameter being the efficiency product ($\eta c\ \eta t\ \eta m$), which is illustrated on the example Fig 3 for three selected rotational speeds 140, 100, and 70 krpm, over an airflow range of 0 to 125 ppm, and pressure ratio of 3.0. The overall T/C peak efficiency occurs at Nsc close to 0.8 but in the interests of lower costs, designs are often biased towards Nsc 1.0 to1.2. This is justifiable since the effects of overall turbocharger efficiency on engine thermal efficiency are far less than the effects of compressor and turbine efficiency on gas turbine cycles, and can be as low as 0.1% for each 1% point gain in ($\eta c*\eta t*\eta m$). Nevertheless improved overall efficiency benefits turbocharger acceleration lag, as the torque available for acceleration at constant turbine inlet temperature can increase 5% for a 1% point gain in ($\eta c* \eta t* \eta m$).

Compressor surge margin can however be more important than thermal efficiency if it is necessary to wastegate during engine operation. Nevertheless improved overall efficiency benefits T/C acceleration lag, as the torque available for acceleration can increase 5% for a 1% point gain in ($\eta c\ \eta t\ \eta m$). An can increase 5% for a 1% point gain in ($\eta c\ \eta t\ \eta m$).

An evaluation of turbocharger mechanical losses (ηm) is presented by Payri et al [3].
The projected variation of turbocharger relative cost and overall efficiency ($\eta c\ \eta t\ \eta m$) is shown on Fig 4 versus compressor specific speed Nsc ,where for example decreasing Nsc from 1.1 to 0.7 doubles relative cost, but increases overall turbocharger efficiency from 41 to 51 %. The trends exemplify the importance of specific speed choice in small turbocharger design where compressor and turbine efficiency are often traded in favor of manufacturing cost, contrary to the majority of microturbine applications.

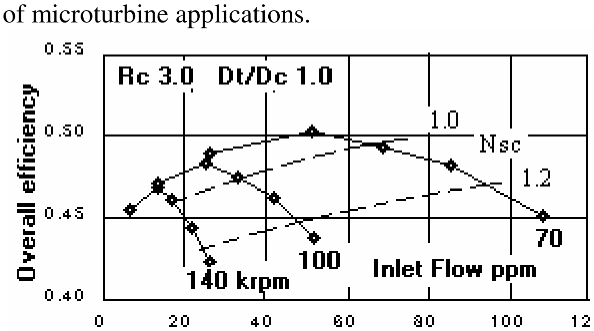

Fig 3. Optimum Small Turbocharger Sizing.

The singular major difference between the small T/C and gas turbine radial inflow turbine is nonwithstanding :

. **The T/C turbine only supplies power to drive its compressor, whereas the gas turbine must additionally provide the net power output.**

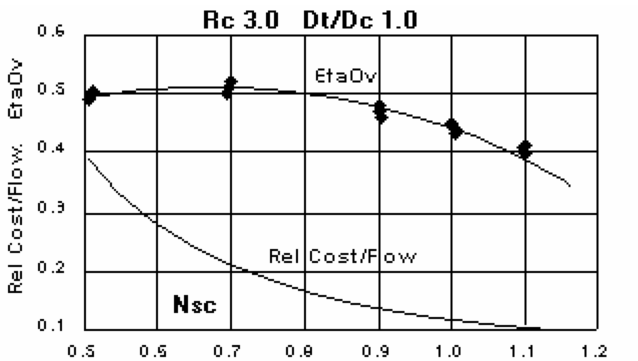

Fig 4. Relative T/C Cost and Efficiency.

As a consequence it can be shown that if microturbines are to operate near maximum efficiency, the ratio of the radial turbine rotor tip diameter to that of the backswept type centrifugal compressor will be required to be larger than that of a T/C. Increasing radial inflow turbine pressure ratios and TIT's above approximately 5.0 and 1850F can encroach a turbine tip speed limit as restricted by rotor stress limitations.

The reciprocating engine pulsating exhaust flow swallowed into the T/C necessitates fairly robust turbine blading, tolerant of hot gas blowdown resonant excitation modes, accordingly blade thicknesses are made stiffer than those of the microturbine imposing further penalties on turbine efficiency. Additionally the thicker trailing edge blade wakes, and tight exducer shroud curvatures from radial to axial directions trigger increased exhaust diffuser entry blockage promoting lower diffuser pressure recovery, if indeed an exhaust diffuser is incorporated at all.

CONVERSION TASKS
- TURBOCHARGER TO MICROTURBINE.

Four major tasks involved in converting a turbocharger into a microturbine are :

1). Addition of the combustor
2). Addition of recuperator for fuel efficiency
3). Development of electronic controls
4). Bearing type selection

2. DESIGN COMBUSTOR.

The design of small gas turbine combustors is somewhat less amenable to computational analysis than the compressor and turbine aerodynamic design due in part to the effects of :

. Surface area / volume increase, with decreasing size.
. Increased effects of wall quenching.
. Low fuel flows necessitating minimum number of injectors and injector orifice sizing.
. Concentricity limitations between the inner and outer liners.
. Increased effect of leakage gaps on emissions and pattern factor

As a consequence there is a reluctance to directly apply scaling from larger combustors and alternative design solutions have been adopted such as the small gas turbine shaft slinger fuel spray and single injector rotating cup fuel atomizer, Rodgers [4]. Cold starting using small injectors with Diesel fuel is especially thought provoking and has been sidetracked in some small microturbine development programs by using natural gas or propane thereby requiring a pressurized gas source. Chen, Mitchell, Nourse [5] successfully developed a microturbine Diesel fueled low emission combustor. Natural gas fumigation at the compressor inlet, Rodgers [6] has been proposed, but flame back particularly in the hot recuperator could be an inherent danger.

Single can combustors have proven popular in most T/C conversions as they can be conveniently integrated in-between the separate compressor and turbine scrolls, but consume additional packaging volume, with slightly increased pressure drop. The dominant combustor design consideration is by far the engine exhaust emissions.

Emissions are rapidly becoming the dominant criterion in the design of small microturbines, to the point that the whole engine design may be focused upon the combustor environment and operation. Fig 5 portrays two conceptual (non converted T/C) microturbine combustor configurations, one with a "reverse" flowpath focused upon fuel injector access and flow uniformity into a single centerline can combustor.

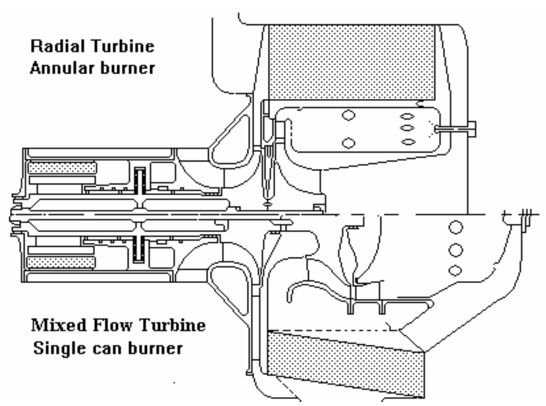

Fig 5. Conceptual Microturbine Annular Combustors.

The vital endeavor to meet new emission requirements will be at part load conditions. The T/C steady state work balance equation (1) can be also used to assess Brayton cycle self-sustaining cycle temperature ratio TIT/T1 and thus the trends of self sustaining fuel/air ratios which reveal that the combustor must possess the ability to operate very lean, but yet with low emissions and high combustor efficiency.

Partload fuel flows can be reduced by decreasing speed either with a two or single shaft engine although this may tend to retard load response. For a recuperated engine speed reduction and hence pressure ratio may eventually incur encroaching a recuperator matrix temperature limit, as dependent upon the initial selection of the design point pressure ratio.

Emission requirements at part load conditions can be accomplished with either a multiple injectors or variable combustor geometry. With fuel staging all of the injectors are used when the engine is at or near full load. As power requirements are reduced an increasing number of injectors are shut off. This allows the temperature of the primary zone of the active injectors to stay at our near their design level thus maintaining the engines low emissions signature. As an example the effects of recuperator effectiveness and cycle pressure ratio with an EGT of 1250F on relative part load emissions are shown on Fig 6, and Fig 7 respectively, indicating that a pressure ratio of 4.0 and effectiveness of 85 % could be a good compromise between NOx and CO.

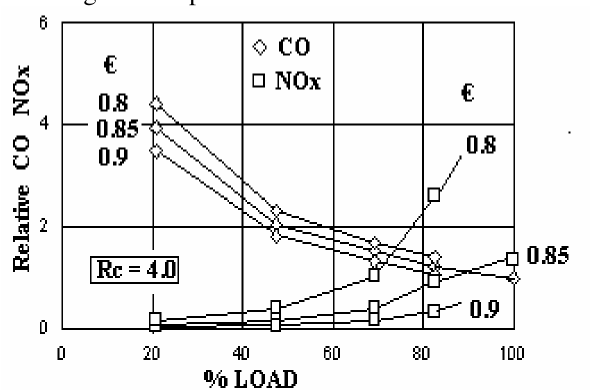

Fig 6. Emission Trends vis Effectiveness.

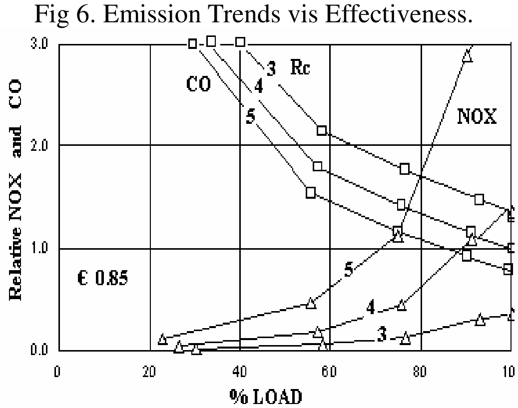

Fig 7. Emission Trends vis Pressure Ratio.

Regardless of which type of combustor is selected, the main requirement will be to produce the required firing temperature at an emissions level that meets or is below the upcoming requirements on a variety of liquid fuels. It is anticipated that these emission levels will not be easy to meet. A lean pre-vaporized, premixed combustor design has the best chance of meeting these requirements. Variable combustor geometry or staging between multiple injectors will be required to keep the emissions low during part load operating conditions. Because of the difficulty in meeting both the low NOx and the low CO emissions requirements, high volume primary and secondary zones are needed. However, some size limits must be placed on the combustor design so it does not adversely affect the overall engine envelope or the engine cost.

4. ADD RECUPERATOR.

Exhaust heat recovery would appear to be a must for continuous duty generator set (genset) applications plausibly with an add-on box type fixed boundary thin foil metallic recuperator, if a converted T/C. Both primary-surface and plate-fin types of construction have been utilized in first-generation microturbines. The former has found increasing acceptance because it is more amenable to automated fabrication and offers a lower cost solution McDonald. C.F, [7].

Recuperator temperature limits are based on the magnitude of the material's tensile strength and its resistance to corrosion, oxidation and creep deformation.. Stainless steel Type 347 is used in many recuperators, but to achieve an acceptable life, a maximum allowable temperature of 1200 – 1300 F limits the turbogenerator efficiency to about 30%.The benefits of exhaust recuperation for automobile battery charging applications would depend upon charge time versus recuperator additional weight, volume, reliability, and cost.

Low weight is not as crucial as cost for a genset applications but weight is the cornerstone of a fuel efficient vehicle since the amount of energy required to accelerate the car is directly proportional to its mass. Light weight for aircraft auxiliary power units (APU's) is so vital that cast components are only utilized where weight and strength equivalence to fabricated or machined components is possible.

Estimated two hour duration wet weights for a 5.0 Kw lightweight microturbine based on aerospace APU technology and manufacturing techniques are shown on Fig 8 at sea level 85F, EGT of 1250F, with parameters of recuperator effectiveness and compressor pressure ratio. The weights assume the batteries are being charged at full load and exclude engine controls (assumed integrated in the vehicle controller) and installation ducting.

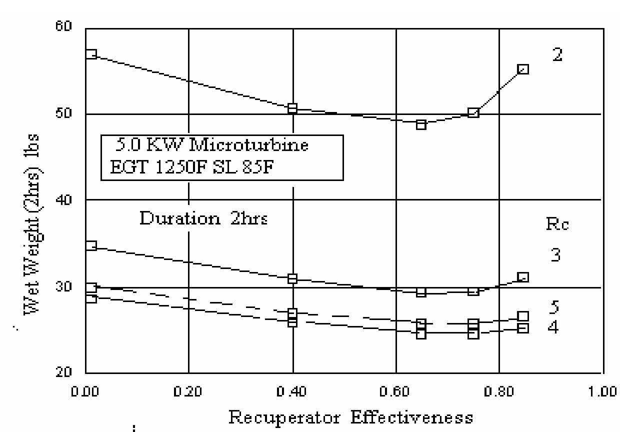

Fig 8. Weight Tradeoff.

These ambitious "aerospace type" weight predictions indicate that recuperation affords a minor weight advantage which could be readily lost in the viability of the weight assessment , Christodoulou .F., et al [8], added to which recuperation increases installation complexity but would help however in reducing exhaust stack exit temperature.
Overall cost and reliability are additional factors which would favor choice of a non recuperated microturbine for battery charging applications, and for even shorter operating durations the option of a range anxiety backup battery emerges.

5 . PROVIDE CONTROLS

The design selection of the engine accessories and type of control hydro-mechanical ,electronic, or hybrid is an extensive topic in itself , suffice it here to qualify the costs as amounting to at least 30% of the complete engine cost depending on the specific design features ,data acquisition, data processing, and transient capabilities. Engine control cost do not follow the scalar relationship, since control dynamic relationships (apart from inertial effects), are relatively independent of size.
The function of the control system is to provide safe operation of the engine under all load and environmental conditions, and to perform consistent starting and shutdown.
Once combustion is fired, the magnitude of engine starting is governed by:

- Engine performance and temperature limits
- Compressor surge margin
- Fuel control scheduling
- Parasitic and applied torques
- Transient thermal mass affects (significant for recuperated engines).

Accessory costs, fuel /oil pumps, starters, electronic equipment and the like are highly dependent upon availability of commercial components suitable for the application. As a consequence engine accessory and control costs tend to remain nearly constant, throughout the power bracket of 5 to 50 Kw output.
The sophistication of these controllers is now being paralleled for variable geometry turbochargers to regulate emissions via exhaust gas recirculation during all vehicle operating modes and environments.

The advent of electronic controls for small gas turbines has permitted the utilization of closed loop acceleration fuel control scheduling where fuel may be topped as a function of a single or multiple control variable. Electronic controllers are also normally programmed to acquiesce and record engine operational data for built in test capability.
Most microturbines provide higher starting torques by metering fuel close to the design transient TIT limitations. Typically, two fast response thermocouples are positioned in the exhaust to measure exhaust gas temperature (EGT) which is a speed dependent indicator of TIT. A suite of protected sensors may include speed pickups, event counter, thermocouples, pressure transducers, and can with the electronic controller represent an appreciable fraction of the total engine cost.

6 . SELECT BEARING TYPE.

During the preliminary mechanical design of the 10 turboalternator Rodgers.C., (Ref 4) for the U.SArmy , an extensive survey of low cost high speed bearings was conducted, both of the conventional rolling element and floating sleeve fluid film types. Of paramount importance in the Army application was cold starting the turbo-alternator at -40 degree C conditions.
Actual bearing tests were conducted to assess the capability of accomplishing cold starting with turbocharger floating sleeve bearings and it was subsequently concluded that undue battery capacity would be required, as compared to the lower viscous drag characteristics of ball bearings. As a result ball bearings were selected.
The drag characteristics of turbocharger fluid film bearings McInnes,Johnson, [9] , reveal that that at rotational speeds of the order 100 krpm power losses of 0.9, and 1.3 , have been documented on 100 and 150 mm turbochargers respectively at temperate ambient day conditions. For the 10 Kw turboalternator with a rated speed of 93.5 krpm this would have amounted to some 10% of the output power,(equivalent to approximately 3% points on turbine efficiency), excluding reduction gearbox and accessory drive losses.
The friction losses in the sleeve bearings and in the thrust bearing can result in mechanical efficiencies in the 70% range in small T/Cs. In one T/C test instance the author witnessed an overall turbine efficiency including sleeve bearings of 58% at 220 krpm.

Small turbo-chargers are now available with either hybrid bearings, or ball bearings with less than half the loss of sleeve bearings. There is little or no oil film shear in antifriction ball bearings which operate with rolling friction only, thus ensuing in higher mechanical efficiencies and faster acceleration rates.
Compared with steel balls, ceramic balls in ball bearings have a number of advantages. Bearing service life is two to five times longer. They run at lower operating temperatures and allow running speeds to be as much as 50% higher. The surface finish of ceramic balls is almost smooth, producing lower friction losses and lower vibration levels. There is less heat buildup during high speed operation, they exhibit reduced ball skidding and have a longer fatigue life. All these characteristics make ceramic ball bearings ideal for use in small T/Cs where they must operate at very high speeds and survive in the higher temperature locations of microturbines.

The primary advantage of air bearings is that no oil system and cooling are required, yet a vehicle entirely devoid of an oil system is hard to perceive. Microturbines with air bearings have exhibited remarkably low self sustaining and brake-away speeds, but are prone to thermal distortion DellaCorte.C., Bruckner, R. [10] and require thrust face diameters almost as large as the compressor tip.

7. MICROTURBINE APPLICATIONS.

The primary application of microturbines to date has been in gensets, some with combined heating and power, imminent applications are being investigated for the anticipated huge hybrid and electric car market as partially addressed herein. Associated applications for small turbomachiney technology are in turbochargers, aviation gas turbine auxiliary power units, unmanned aircraft, closed Brayton cycle units Space power, and predominantly aviation environmental control units.

7.1 AUTOMOBILE BATTERY CHARGER.

For automobile application high start reliability would be a key design factor, ostensibly more so than fuel economy while still satisfying emission regulations. Start reliability at sub zero temperatures and altitudes up to Pikes peak together with clean emissions could be a challenge for a small combustor which would also need its own highly reliable ignition system, but higher combustor equivalence ratios than a recuperated engine, no recuperator matrix heat warm up period, plus lower bearing viscous drag than a reciprocating engine are factors conducive to starting ease.

Probable power range would be 5-20 Kw with ability to meet starts at underhood conditions during hot (100F) days. A host small T/Cs in the 150-250 krpm range could be candidates for this battery charging range extender application which converted into low cost non recuperated microturbines would have hot day (85F) thermal efficiencies in the 10% range.

7.2 CONTINOUS DUTY GENERATOR SET.

Representative thermodynamic performances of recuperated cycle microturbines, and converted T/C microturbines for generator sets and small domestic combined heat and power units are depicted on Fig 8, where thermal efficiency, ETATH, and specific power are plotted versus pressure ratio for either a TIT of 1850F or EGT of 1250F whichever occurs first as limited by life expectancy of the turbine rotor superalloy or recuperator matrix alloy.

These trends were based upon converted T/C compressor and turbine efficiency decrements of 1.5% points and 6.0% points respectively plus 1% additional duct pressure loss as compared to existing microturbine component performance levels. Although retro fitting a converted T/C with a higher performance turbine might mitigate some of the performance difference a TIT of 1650F could be unmanageable . A side effect too is that some 30% larger recuperator would be required to attain the same power as for a fixed effectiveness.

Empirical emissions relationships for NOx and CO levels at the design point cycle conditions studied showed relative emission levels to double without recuperation with the higher fuel flows and only a minor advantage from increased pressure ratios.

Fig 8 illustrates that specific power almost doubles increasing pressure ratio from 2.0 to 4.0.

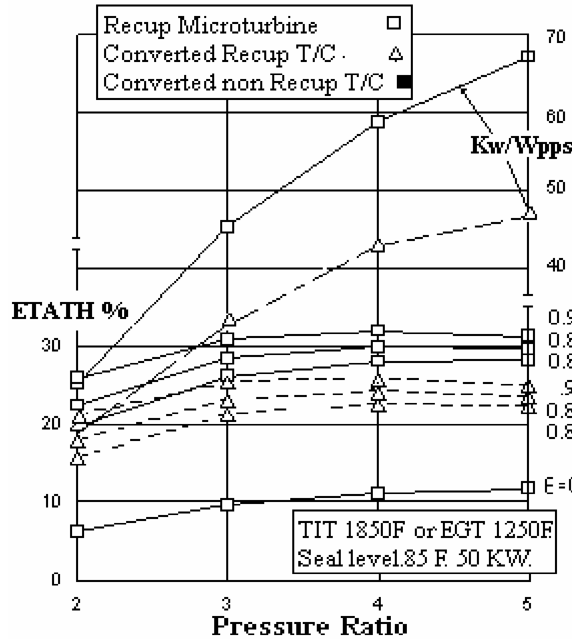

Fig 8. Recuperated Microturbine Thermal Efficiencies.

This is significant in that engine size and dry weight is dependent upon specific power (Kw/W) yet fuel weight for a given trip is primarily dependent on SFC and load profile. It can be shown for weight dominated vehicles that a parameter representative of relative duration is :

$$\text{Range} \quad fn \quad (\text{Kw} / \text{W})^{0.5} \; / \; (\text{SFC})$$

Such a range parameter can be effectively utilized in preliminary design to optimize the choice between the trends of power density and specific fuel consumption as reflected in Fig 9 where relative range is plotted versus pressure ratio with recuperator effectiveness as a parameter substantiating an optimum pressure ratio of 4.0.

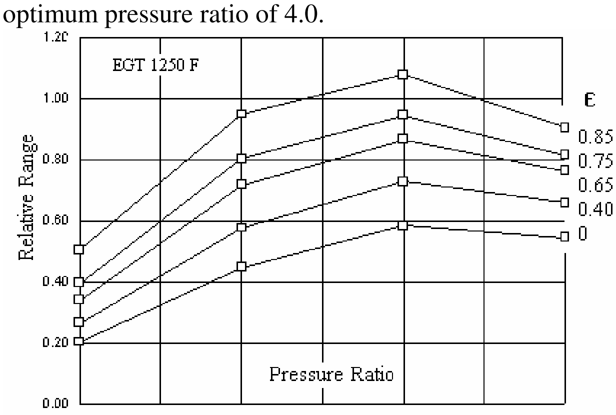

Fig 9. Range Optimization Trends

8. COST EVALUATION.

The prime consideration in any T/C to microturbine convertibility program is the developed product potential to successfully compete in the open market against existing Brayton, Otto, and Diesel engines.

It is clear that the non-recuperated cycle configuration would have difficulty competing on continuous duty operating cost basis unless coupled with some form of waste heat recovery. Standby operation such as a battery charger for a hybrid car as discussed could be huge market enticement.

Major cost contributors are the electronic controls, sensors, accessories, frequency converter for the genset high speed PMG and recuperator. An approximate component cost breakdown for a recuperated microturbine McDonald. C.F., Rodgers.C. [11] is shown on the pie chart Fig 10 which illustrates that powerhead module constitutes approximately one third of the total engine cost, and that although powerhead manufacturing cost could be size dependent similar to mass produced T/Cs, the associated ancillary component costs (such as e.g., the control system) could be expected to remain almost fixed independent of size in the relevant power range. Regardless of manufacturing cost projections the whole concept of cost reduction via T/C conversion may be an illusion since the powerhead cost of a converted turbocharger microturbine embodying all the improvements described may well equal that of a indvidually designed microturbine, furthermore this would constrain the recuperator to the "add on" type only.

The cycle analysis code used in this treatise was consequently extended to include relative cost estimates encompassing constant ancillary costs which revealed that projected overall genset specific cost \$/Kw could increase with diminishing engine power output in the manner shown on Fig 11.

The message then becomes clear that converting a turbocharger into a gas turbine essentially provides minimal overall cost advantage, unless the costs of the ancillary equipment can also be reduced, a goal rigorously exercised by competitive small engine manufacturers

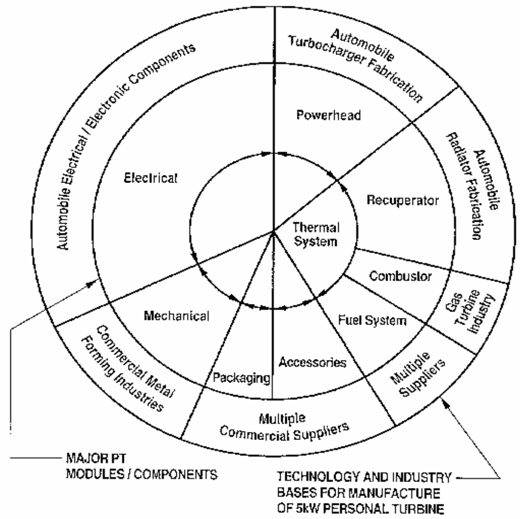

Fig 10. Cost Breakdown.

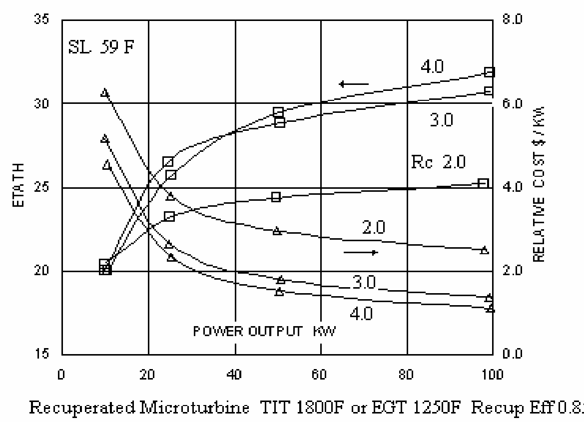

Fig 11. Genset Relative Cost Trends.

Table 3. Modifications to Turbocharger for Microturbine Application

. Compressor	Install vaned diffuser. Assess need for inducer shroud bleed
. Turbine	Incorporate larger diameter rotor, vaned nozzle and exhaust diffuser
. Combustor	Design and develop combustor to meet impending emissions.
Rotating Assembly	Provide front drive power output coupling (as compressor hub size permits). Re-assess shaft dynamics and engine axial end thrust
. Accessories	Provide electric driven accessories and oil/fuel pump
. Controls	Provide electronic control unit, speed governor, overspeed limiter, over- temperature protection and start sequence logic
. Inlet and Exhaust	Inlet filter, and exhaust stack
. Test	Shakedown and final quality acceptance test

9. CONCLUSIONS.

The primary purpose of this paper was to examine the viability of achieving lower cost small gas turbines by the concept of turbocharger convertibility. It has been shown that if competitive performance to existing single recuperated shaft microturbines is to be attained, at least the modifications and component developments listed above in Table 3 should be addressed:
Some of these suggested modifications might be accomplished a-priori as a large venue of turbocharger components are often available in a single frame size .
Some immediate preferences might conveniently be the largest diameter turbine, ceramic antifriction bearings, no wastegate and possibly vaned diffuser and vaned turbine nozzle.
Since essentially one third of microturbine overall costs are absorbed in the turbomachinery, the net cost savings adapting a suitable off-the shelf turbocharger could be marginal, exclusive of increased operating costs as a continuous duty genset.
In essence the ancillary costs could far outweigh the relative importance of the powerhead module and operational costs would clearly favor recuperated engines with the highest component efficiencies. If battery charge periods took several hours there could be additional substantiation for recuperation.
The turbocharger itself is no longer that simple add-on power boosting module and is evolving into a sophisticated variable geometry machine needed to regulate emissions via exhaust gas recirculation during all vehicle operating modes and environments.

If small gas turbines are to compete with Diesel and Otto cycle engines it may eventually be in emissions sensitive applications, or possibly through breakthroughs in the development of ceramic [12] and composite materials. Empirical emissions computations at design point conditions showed that a cycle pressure ratio of 4.0 with a recuperator effectiveness of 85 % could offers both a good compromise between NOx and CO plus continuous duty fuel costs, but the major concern could irrevocably become emissions during part load and transient operating conditions.
It should be emphasized that the cycle performance conclusions are based on either a TIT of 1850F or EGT of 1250F whichever occurs first, typical of temperature limits for current turbine superalloys and stainless steel recuperator foil.

The conversion of T/Cs to small gas turbines has been demonstrated on many occasions in the past particularly in engineering learning institutions, but as yet the convertibility route discussed herein appears only an exploratory sidetrack to a perceivable viable microturbine marketplace. Turbocharger manufacturers are exploring such a route, Ryder et al., [13]. [14], but as yet only cautiously exploited the commercial challenge. A dual function combined turbocharger and auxiliary power unit [15] McDonald. C.F., Rodgers.C. may be the eventual candidate.

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