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DISSERTATION

ON

FEASIBILITY STUDY ON FITMENT OF GAS TURBINE ON TANK T-72

BY
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GUIDE
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FEASIBILITY STUDY ON FITMENT OF GAS TURBINE IN TANK T-72

BY

MAJOR M.RAVI

DISSERTATION

SUBMITTED IN PARTIAL FULFILMENT FOR THE AWARD
OF DEGREE OF
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(MECHANICAL COMBAT VEHICLES)
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CERTIFICATE

CERTIFIED THAT THE DISSERTATION ON "FEASIBILITY STUDY ON FITMENT OF GAS TURBINE IN TANK T-72" SUBMITTED BY MAJOR M RAVI, EME IN PARTIAL FULFILMENT OF THE REQUIREMENT FOR THE AWARD OF DEGREE OF MASTER OF TECHNOLOGY (MECHANICAL COMBAT VEHICLES) OF JAWAHARLAL NEHRU UNIVERSITY OF NEW DELHI, IS A RECORD THE STUDENTS OWN WORK CARRIED OUT UNDER MY SUPERVISION AND GUIDANCE.

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INTRODUCTION

1. Of the various means of producing mechanical power the turbine is in many respects most satisfactory, the absence of reciprocating members the balancing problems are few, the lubricating oil consumption is exceptionally low and the reliability can be high. Also if future uprating is kept in mind at design stage it self the gas turbine power can be increased to as much as three times by such methods as increasing the mass flow, turbine inlet temperature, component efficiencies etc while maintaining the same basic engine design.

2. It is these advantages of gas turbines that the military planners world over thought of adopting the gas turbine to Armoured fighting vehicles also. Engine of an AFV is one of the basic contributing factor towards mobility. The charecteristic of gas turbine to uprate the engine power with out changing the basic design is the most attractive lure towards AFV applications, mainly because the improvements on mobility, protection and fire power can be made on to the AFV with out having to go through the entire design procedure. This also enhances the life of the AFV and saving in cost. The countries like U.K., France, Germany and USSR tried gas turbines in AFVS. U.S. has successfully fitted gas turbine in XMI which is presently in service with U.S. Army and Marines and also few of them have been exported to Saudi Arabia and Egypt.

AIM

4. To study and examine the feasibility of fitment of Gas turbine in TK T-72.

5. The paper will be covered in following Phases.

PHASES

(a) Working of simple gas turbine.

(b) Types of Gas turbines.

(c) Advantages and disadvantages of gas turbines.

(d) Why gas turbines for Tanks.

(e) Design procedure of gas turbine.

(f) Volumetric analysis of Engine compartment of Tank T-72.

(g) V-46-6 Engine specifications.

(h) Gas turbine that are available in the world.

(j) Selection of suitable gas turbine for fitment.

(k) Design of air cleaner.

(l) Design of reduction gear box.

(m) Further advantages of gas turbines.

(n) Comments on feasibility of fitment.

(o) Latest developments in the field of gas turbines for tank applications.

(p) Conclusion

WORKING OF SIMPLE GAS TURBINE

5. A simple gas turbine is as shown in Fig-1.

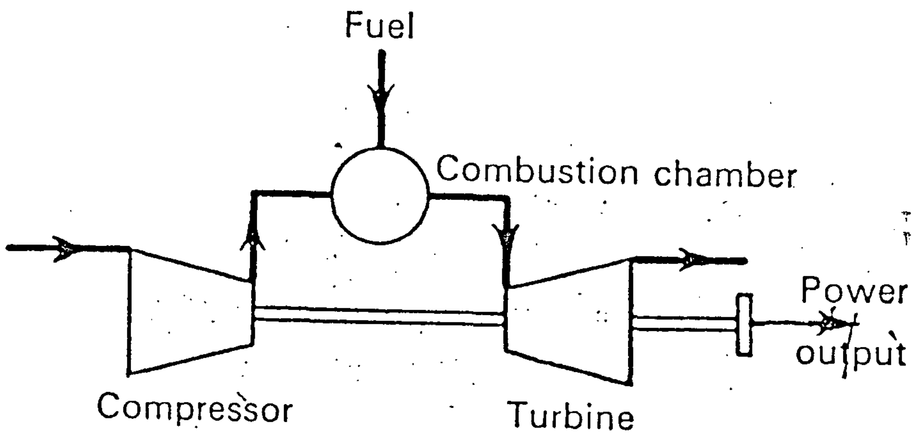


Fig-1.

A gas turbine consists of compressor, combustion chamber, turbine and the control system. In order to produce an expansion through a turbine a pressure ratio must be provided and the first necessary step in the cycle of a gas turbine plant must therefore be compression of working fluid. If after compression the working fluid were to be expanded directly in the turbine and there were no losses in either component the power developed by the turbine would just equal that absorbed by the compressor. Thus if the two were coupled together the combination would do no more than turn itself round. But the power developed by the turbine can be increased by addition of energy to raise the temperature of the working fluid prior to expansion. When the working fluid is air the most suitable means of doing this is by combustion of fuel in air which has been compressed. This process takes place at combustion chamber after the fuel is sprayed and ignited. The hot working fluid through its passage on turbine expands because of which some work is done on the turbine blades due to which the turbine rotates. Here the part of the work is used up in driving the compressor and part of it is available to do some other external work if coupled to a load. In practice losses occur at both compressor and turbine which increases the power absorbed by the compressor and decreases the power output of the turbine.

TYPES OF GAS TURBINE

6. There are two types of gas turbine, they are :

   (a) open cycle gas turbine.

   (b) closed cycle gas turbine.

7. **Open Cycle Gas Turbine.** Diagramatically an open cycle gas turbine is as shown in the fig-1 : Here fresh atmospheric air is drawn into the circuit continously and energy is added by the combustion of fuel in the working fluid itself. In this case the products of combustion are expanded through the turbine and exhausted to atmosphere.

8. **Closed cycle gas turbine.** A closed cycle gas turbine is diagramatically represented at Fig-2.

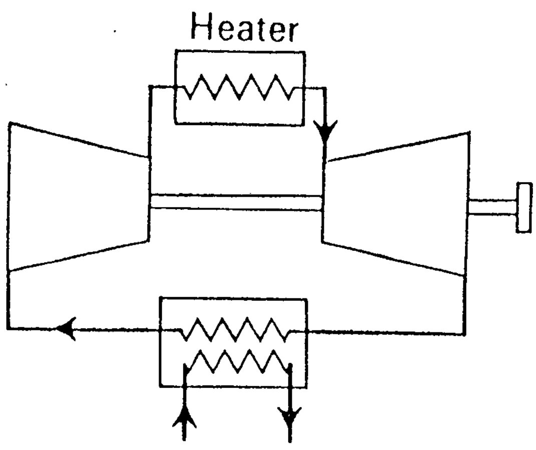


Fig-2.

Here the working fluid is repeatedly circulated through the machine. It is clear in this type of plant the fuel can not be burnt in the working fluid and the necessary energy must be added in a heater or "gas boiler" where in the fuel is burnt in a separate air stream supplied by an auxiliary fan. Though numerous advantages are claimed in case of a closed cycle gas turbine they are more suitable for static applications particularly nuclear reactor where the products are dangerous if exhausted to atmosphere because of the possibility of radioactive contamination.

9. Closed cycle gas is quite complex bulky and costly as compared to that of open cycle gas turbine. The construction is required to be gas tight, requires large amount of cooling water for precooler, a heavy and large heater and quite a complex system. This finds application only in static & Industrial and marines. They are not used in aviation or land application for this reason and hence no more discussed.

OPEN CYCLE GAS TURBINE

10. Further classified into

(a) Constant Pressure Cycle.

(b) Constant Volume Cycle.

(a) Constant Pressure Cycle. Also known as Joule (or Brayton) Cycle. The P-V & T-S diagram is shown in the figure-3.

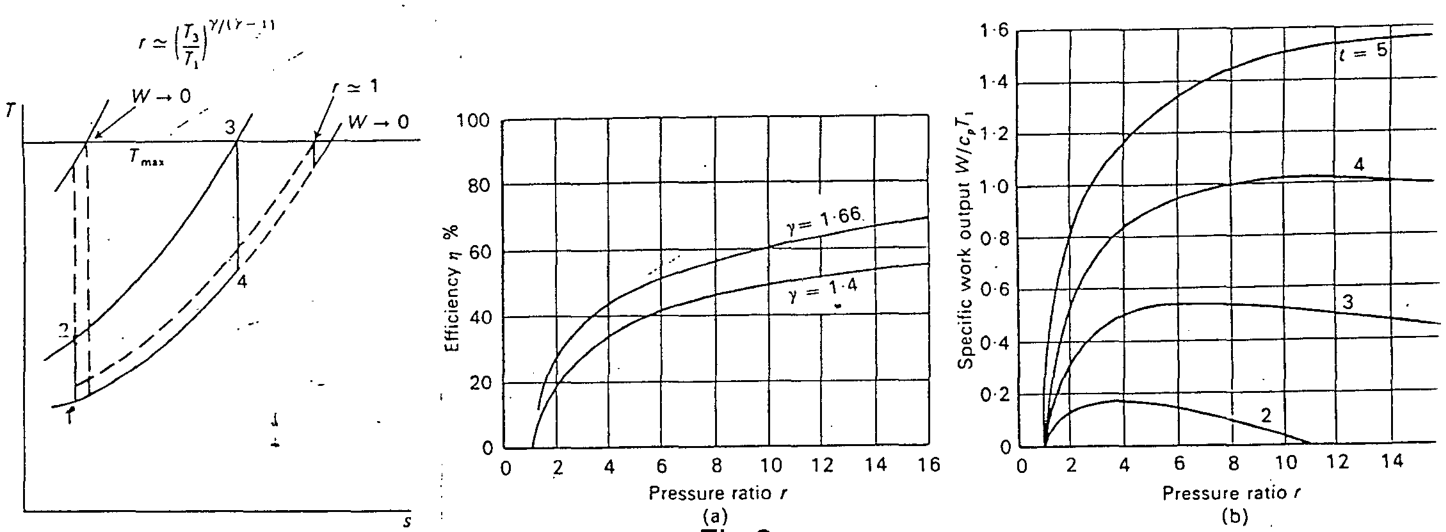


Fig-3.

Following assumptions are made analysing the cycle.

(i) Compression and expansion process are reversible and adiabatic i.e. isentropic.

(ii) The change of kinetic energy of the working fluid between inlet and outlet of each component is negligible.

(iii) There are no pressure losses in the inlet ducting, combustion chambers, heat exchangers, intercoolers, exhaust ducting, and ducts connecting the components.

(iv) The working fluid has the same composition through out the cycle and is a perfect gas with constant specific heats.

(v) The mass flow of gas is constant through out the cycle.

(vi) Heat transfer in a heat exchanger is `Complete', so i.e. in conjunction with (iv) & (v) the temperature rise on the cold side is max possible and exactly equal to the temperature drop on the hot side.

Steady state flow energy equation is

$$Q = (h_2 - h_1) + 1/2\,(C_2^2 - C_1^2) + W$$

Where Q and W are the heat and work transfer per unit mass flow. Applying to each component, bearing assumption (ii) in mind.

$$W12 = -(h_2 - h_1) = -Cp\,(T_2 - T_1)$$

$$Q23 = (h_3 - h_2) = Cp\ (T_3 - T_2)$$

$$W24 = (h_3 - h_4) = Cp\ (T_3 - T_4)$$

The cycle efficiency is

$$\eta = \frac{\text{net workoutput}}{\text{heat supplied}}$$

$$= \frac{Cp\ (T_3 - T_4) - Cp\ (T_2 - T_1)}{Cp\ (T_3 - T_2)}$$

Making use of isentropic P-t relation

$$T_2/T_1 = r^{(r-1)/r} = T_3/T_4$$

where $r = P_2/P_1 = P_3/P_4$

$$\eta = 1 - (1/r)^{(r-1)/r} \qquad (1)$$

11. Constant volume cycle

Atkinson cycle.

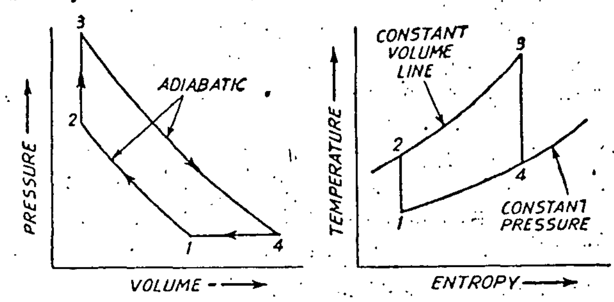


Fig-4.

heat supplied = $Cv\ (T_3 - T_2)$
heat rejected = $Cp\ (T_4 - T_1)$

$$\eta_{\text{overhaul}} = \frac{\text{Heat supplied - Heat rejected}}{\text{Heat supplied}}$$

$$= \frac{Cv\ (T_3 - T_2) - Cp\ (T_4 - T_1)}{Cv\ (T_3 - T_2)}$$

$$= 1 - cp/cv \frac{T_4 - T_1}{T_3 - T_2}$$

making use of P - T relation Cp/Cv = r

$$\eta_{\text{overhaul}} = 1 - \frac{\gamma(r-\alpha)}{(r^\gamma - \alpha^\gamma)} \quad \text{\_\_\_\_\_\_\_\_\_\_\_\_} (2)$$

12. Comparing both constant pressure and constant volume cycle.

(a) The max pressure is well above the compression pressure, which means that this cycle needs maximum number of high pressure blades as compared to that of constant pressure turbine.

(b) The design is complicated as the constant volume cycle requires valves in the combustion chamber.

(c) Incase of constant volume cycle flow is intermittent as compared to constant pressure cycle where flow is continous.

13. The efficiency not only depends on pressure ratio and type of gas but also of maximum cycle temperature T_3

$$W = Cp\,(T_3 - T_4) - Cp\,(T_2 - T_1)$$

This can be expressed as

$$W/CpT_1 = t\,(1 - 1/\eta^{(r-1)/r}) - (r^{(r-1)/r)} - 1) \text{\_\_\_\_\_\_} (3)$$

Where $t = T_3/T_1$, where T_1 is normally atmospheric temperature and is not a major independent variable. A plot of specific work output in non dimentional for ($w/Cp\,T_1$) as a function of r and t is shown in fig. The value of T_3 which is the maximum cycle temperature and hence t1, that can be with stood by the highly stressed turbine parts for required working life which is also common as metallurgical limit.

For industrial plant it is 3.5 to 4, for aircrafts 5 to 5.5 for autos it may be between these two. From the T-S diagram W = o when r = 1 and at value of r of which the compression and expansion coincide, i.e. $r = t^{r/(r-1)}$ for any given value of `t' the optimum pressure ratio for maximum specific work output can be found by differantiating equation - (3) and equating to zero.

$$\eta_{opt}^{(r-1)/r} = \sqrt{t}$$

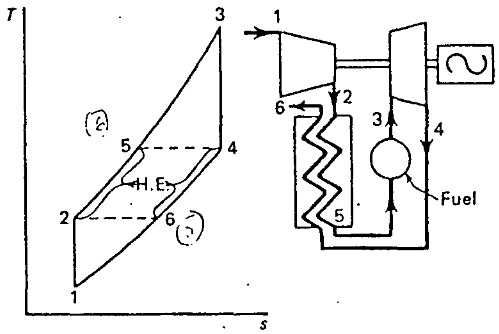
since $r^{(\gamma-1)/\gamma} = T_2/T_1 = T_3/T_4$

$T_2/T_1 \times T_3/T_4 = t$

But $t = T_3/T_1$ & consequently follows $T_2 = T_4$

The specific work output is maximum when the pressure ratio is such that the compressor turbine outlet temperature are equal for all values of r between 1 & $t^{r/2(r-1)}$, T_4 will be greater than T_2 and heat exchanger can be incoporated to reduce the heat transfer from the external source and hence increase in efficiency

14. Heat exchange cycle.



Simple cycle with heat-exchange

Fig-5.

$$\eta = \frac{Cp\,(T_3 - T_4) - Cp\,(T_2 - T_1)}{Cp\,(T_3 - T_5)}$$

with ideal heat exchange $T_5 - T_4$ and substituting the isentropic P - T relation the expression reduces to

$$\eta = \frac{1-r^{(\gamma-1)/\gamma}}{t}$$

15. The efficiency of the heat exchange cycle is not independent of maximum cycle temperature, and clearly it increases as t is increased. Further more it is evident that, for a given value of t, the efficiency increases with decrease in pressure ratio and not with increase in pressure ratio for a simple cycle as shown in Fig-6

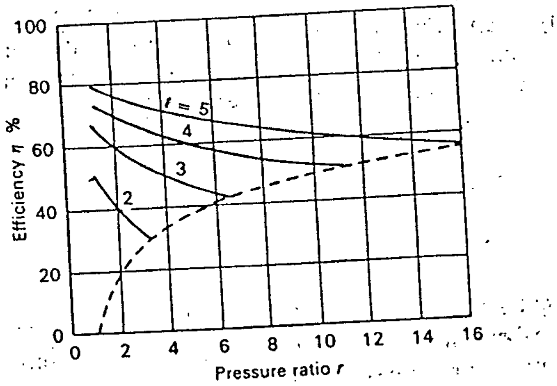


Fig-6.

16. <u>Reheat cycle</u>

A substantial increase in specific work output can be obtained by splitting the expansion and reheating the gas between the high pressure and low pressure turbine. The relevant portion of the reheat cycle on T-s diagram is shown in Fig-7.

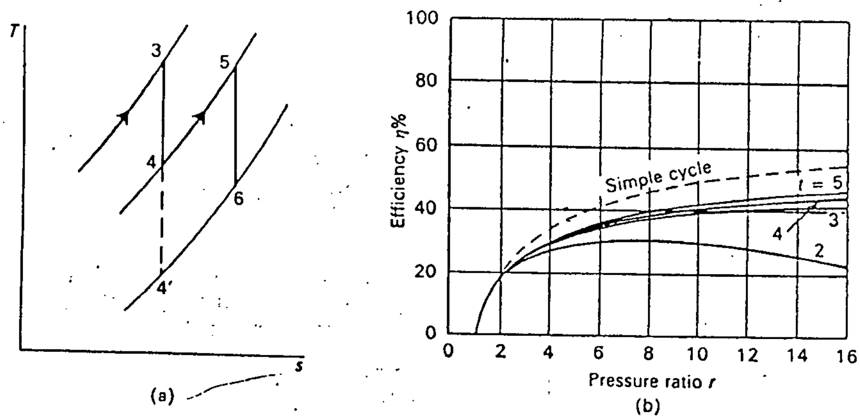


Fig-7.

The turbine work is increased when the vertical distance between any pair of constant pressure lines increases as the entropy increases :

$$\text{Thus } (T_3 - T_4) + (T_5 - T_6) > (T_3 - T_4)$$

17. Assuming that the gas is reheated to a temperature equal to T_3, differentiation of the expression at which to reheat is when the pressure ratio for HP & LP turbine are equal. The expressions for the specific work output and efficiency in terms of r and t

$c = r^{(\gamma-1)/\gamma}$

$W/C_pT_1 = 2t - c + 1 - 2t/\sqrt{c}$

$$\text{efficiency} = \frac{2t - c + 1 - t/\sqrt{c}}{2t - c - t/\sqrt{c}}$$

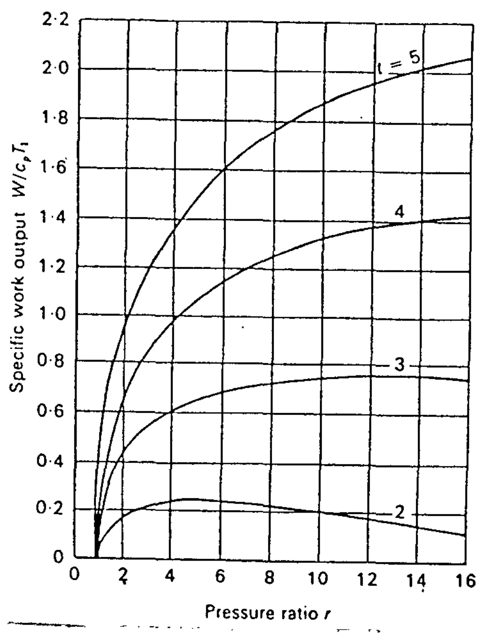
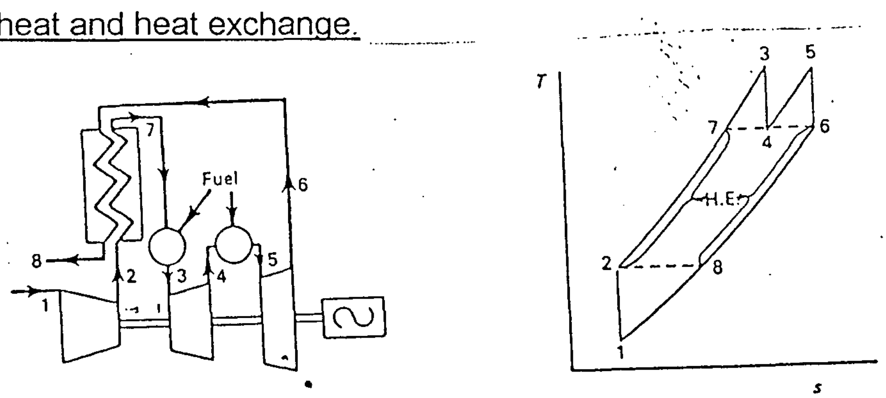


Fig-8.

comparision of $W/C_p T_1$ of fig -8 with that of fig - 3(d) shows that reheat markedly increases specific output from fig -8 (b) indicates this is achieved at the expence of efficiency. However, that the reduction in efficiency becomes less severe as the maximum cycle temperature is increased.

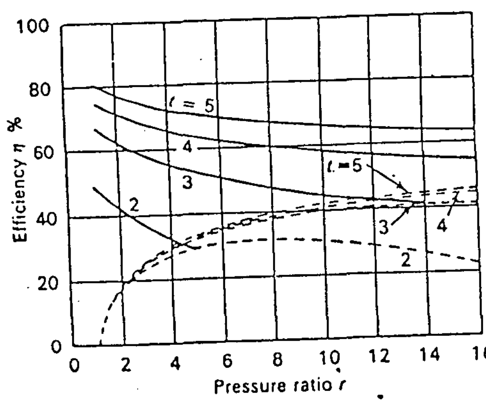
18. <u>Reaheat and heat exchange.</u>



Reheat cycle with heat-exchange

Fig-9.

The reduction in efficiency due to reheat can be overcome by adding heat exchange as shown in fig - 9. The higher exhaust gas temperature is now fully utilised in the heat exchanger and the increase in work output is no longer offset by the increase in heat supplied. But when a heat exchanger is employed, the efficiency is higher with reheat than without as shown in fig 10 & fig 6. The family of const `t' lines exhibit the same features as those for the simple cycle with heat exchange - each curve having the carnot value at $\gamma = 1$ and falling with increasing `γ' to meet the corresponding efficiency curve of the reheat cycle without heat - exchange at the value of `r' corresponding to maximum specific work output.



Efficiency—reheat cycle with heat-exchange

Fig-10.

19. A similar improvement in specific work output to that obtained by reheat can be achieved by splitting the compression and inter cooling the gas between LP & HP compressor; and assuming that the air is intercooled to T1 it can be shown that the specific work output is maximum when the pressure ratios of LP & HP compressors are equal. The use of intercoolers is seldom contemplated in practice because they are bulky and need large quantities of cooling water. The main advantage of gas turbine being compact and self contained is then lost, and hence no more discussed.

20. <u>METHODS OF ACCOUNTING FOR COMPONENT LOSSES</u>

The performance of real cycles differs from that of ideal cycles for the reasons.

(a) Since the velocities of fluids are high in turbomachinery the change in kinetic energy between inlet and outlet of each component can not necessarily be ignored compression and expansion processes are irreversible adiabatics and therefore involve an increase in entropy.

(b) Fluid friction results in pressure losses in combustion chambers and heat exchangers, and also in the inlet and exhaust ducts.

(c) If a heat exchanger is to be of economic size, terminal temperature differences are inevitable.

(d) Slightly more work than that required for the compression process will be necessary to over come bearing and `windage' friction in the transmission between compressor and turbine, and to drive ancillary components such as fuel and oil pumps.

(e) The values of Cp and r of the working fluid vary throughout the cycle due to changes of temperature and, with internal combusiton, due to changes in chemical compositon.

(f) The definition of the efficiency of an ideal cycle is unambiguous, but this is not the case for an open cycle with internal combustion. Knowing the compressor delivery temperature, composition of the fuel, and turbine inlet temperature required, allow for incomplete combustion.

(g) With internal combustion, the mass flow through the turbine might thought to be greater than that through the compressor by virtue of the fuel added. In practice, about 1-2 percent of the compressed air is bled off for cooling turbine discs and blade roots, that the fuel/air ratio employed is in the region 0.01 - 0.02.

21. In para 20 (a) to (f) it has been discussed here factors because of which component losses occur. It is necessary to account for varous component losses as incase of land application particularly Tanks, they have to operate in varied ambient conditions. These problems are necessary so that the choice of turbine can be made correctly.

22. <u>Stagnation properties</u> The kinetic energy terms in the steady flow energy equation by making use of the concept of stagnation enthalpy (total enthalpy) physically, the stagnation enthalpy ho is the enthalpy which a gas stream of enthalpy `h' and velocity c would possess when brought to rest adiabatically and without work transfer. The energy equation reduces to

(ho-h) + 1/2 (0-c2) = 0

Thus ho = h + c2/2

To = T + c2/2Cp incase of perfect gas, h = CpT & stagnation temperature ho = Cp To

$c2/2Cp$ is called the dynamic temperature. It follows from the energy equation that if there is no heat or work transfer To will remain constant. Applying the concept to an adiabatic compression the energy equation becomes

$$W = -Cp(T_2 - T_1) - 1/2(C_2^2 - C_1^2) = -Cp(To2 - To1)$$

$$Q = Cp\ (To2 - To1)$$

23. Stagnation pressure is defined by

$$Po/P = (To/T)^{\gamma/(\gamma-1)}$$

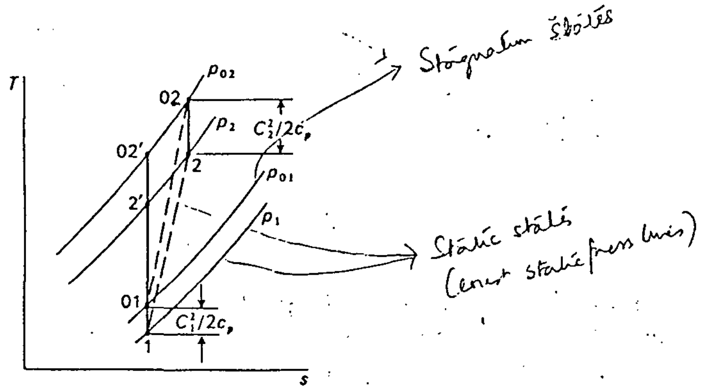


Fig-11.

24. <u>Compressor and Turbine Efficiencies.</u> The efficiency of any machine the object of which is the absorption or production of work, is expressed in terms of the ratio of actual and ideal work transfers. Turbomachines are essentially adiabatic, the ideal process is isentropic and the efficiency is called an isentropic efficiency.

Compressor efficiency = $W^1/w = \Delta ho'/\Delta ho$

$$\eta_c = \frac{To2' - To1}{To2 - To1}$$

similarly

$$\text{Turbine efficiency} = W/W' = \frac{To3 - To4}{To3 - To4'}$$

25. The ideal quantity of turbine work would then seen to be taken more appropriately as that produced by an isentropic expansion from Po3 to the static outlet pressure P4, with P4 equal to the ambient pressure p_a

$$\text{Turbine efficiency} = \frac{T03 - T04}{T03\,[1-(1/P03/Pa)^{(r-1)/r}]}$$

26. In practice a gas turbine is never a single stage both in compressor or turbine. Incase of compressor when the air during its passage is heated up due to friction between air and the compressor blade. Because of this there is an increase in work requirement in the next stage. In the similar manner while the burnt gases when passing over the turbine some heat is gained by the gas due to friction between gas and blades. This increases the energy of the gas leaving that stage. This increase in temperature is regained in the subsequent stage as work. This can be termed as preheat effect. This has resulted in another term called polytropic efficiency and is denoted by (ηp) which is nothing but isentropic efficiency of an elemental stage and is expressed by

For compression

$\eta_{pc} = dT1/dT = \text{const.}$ but $T/P^{(r-1)/1} = \text{const.}$

differntiating and substituting

$$\eta pc = \frac{Ln\,(P2/P1)^{(r-1)/r}}{Ln\,T2/T1}$$

writing above equation

$$T2/T1 = (P2/P1)^{(r-1)/r\eta pc}$$

$$\eta c = \frac{T21/T1 - 1}{T2/T1 - 1} = \frac{(P2/P1)^{(r-1)/r} - 1}{(P2/P1)^{(r-1)/r\eta Pc} - 1}$$

Similarly for Turbine

$$T3/T4 = (P3/P4)^{\eta Pt(r-1)/1}$$

$$\text{efficiency of turbine} = \frac{1 - (1/P3/P4)^{\eta Pt(r-1)/r}}{1 - (1/P3/P4)^{(r-1)/r}}$$

27. The plot of compressor and turbine efficiencies versus pressure ratio is shown on the Fig-12.

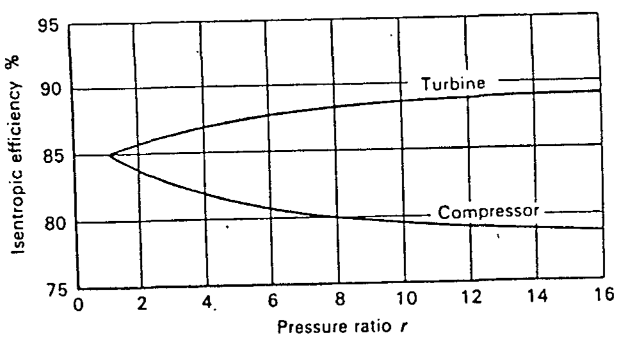
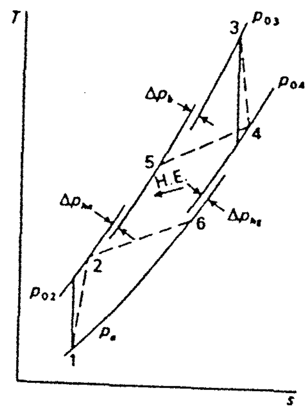


Fig-12.

28. Pressure Losses. It has been amply clear from earlier discussion the pressure Loss occurs at various stages due to windage, friction etc. Similarly a loss in stagnation pressure occurs in combustion chamber due to the aerodynamic resistance of flame stabilizing and mixing devices and also due to momentum changes produced by the exothermic reaction.



Pressure losses

Fig-13.

When a heat exchanger is induced in the plant there will also be frictional pressure losses in the passages on the air-side (Δpha) and gas side (Δphg) as shown in the above fig. A gas turbine is sensitive is irreversibilities, because the net output is the difference of two large quantities and hence the pressure losses have significant effect on the cycle performance.

$$P03 = P02 - Pb - Pha \ \& \ P04 = Pa + P\eta g$$

29. <u>Heat - exchanger effectiveness.</u> The heat exchanger can take many forms, including counter flow and cross-flow recuperators (where hot and cold streams exchange heat through seperating walls) or regenerators (where the streams are brought cydiclly into contact with a matrix which alternately absorbs and rejects heat). In all cases the turbine exhaust gases reject heat at the rate of mb Cp46 (T04 - T06) and compressor delivery air receives heat at the rate of mc Cp 25 (T05 - T02). For conservation of energy, assuming that the mass flow mt and mc are equal.

$$Cp46\,(T04 - T06) = Cp25\,(T05 - T02)$$

The mean specific heat does not vary over the two temperature range then the effectiveness is given by

$$\text{effictiveness} = \frac{T05 - T02}{T04 - T02}$$

30. But in general larger the volume of the heat exchanger the higher can be the effectiveness, but with gas turbines for road or marine transport space is a vital limiting factor, with the development of ceramic materials a compact surface heat exchanger, the effectiveness have increased considerably and have values of effectiveness around 0.9 and can withstand turbine exit temperature of upto 900K. But however, the heat exchangers can not be indefinitely more to have maximum effectiveness incase of Road and Rail application.

31. Mechanical Losses. In all gas turbines, the power necessary to drive the compressor is transmitted directly from the turbine without any intermediate gearing. Any loss that occurs is only due to bearing friction and windage. The loss is very small and is about one percent of the power necessary to drive the compressor. Any power used to drive ancillary components such as fuel oil pumps, can be accounted for by subtracting it from the net output of the unit. Some power is also absorbed in any gearing between the gas turbine and the load.

32. Variation of Specific Heat. The properties Cp and r play an important part in the estimation of cycle performances, and it is necessory to take account of variations in values due to changing conditions through the cycle. This variations invalues due to changing conditions through the cycle is given in the Fig-14.

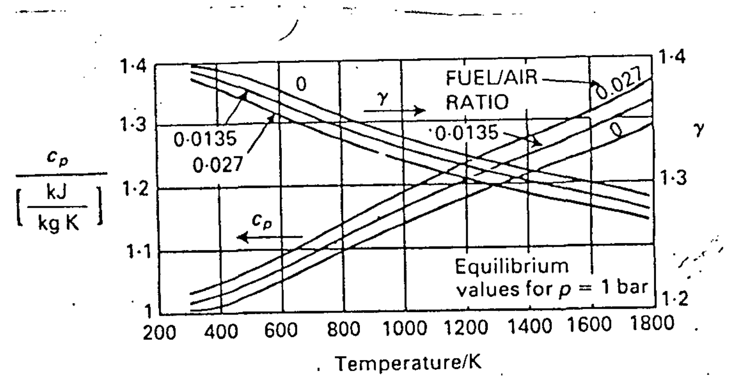


Fig-14.

33. It is also necessary to know the specific fuel consumption of a gas turbine which is one of the main requirement that governs the selection of a gas turbine unless the gas turbine is required to operate on some by product of any major production process.

SFC = f/Wn Kg/KWH.

Where F = Fuel qty in kg/hr.

Wn = KW/kg/s of air flow.

SFC/kg/KWH = F/Wn KWS/Kg X S/n

$$= \frac{3600\ F}{Wn} \text{ (KWS/Kg)}$$

34. The SFC like power output is a function of both compressor speed and power turbine speed. However, it is more convenient to express the SFC as a function of power output. The plot is as shown in the Fig-15.

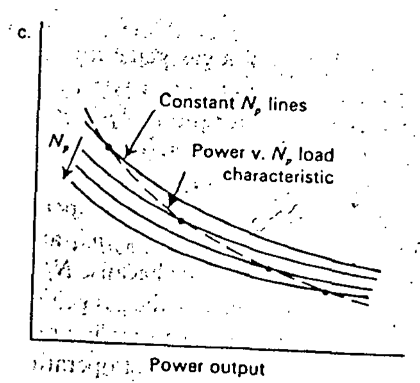
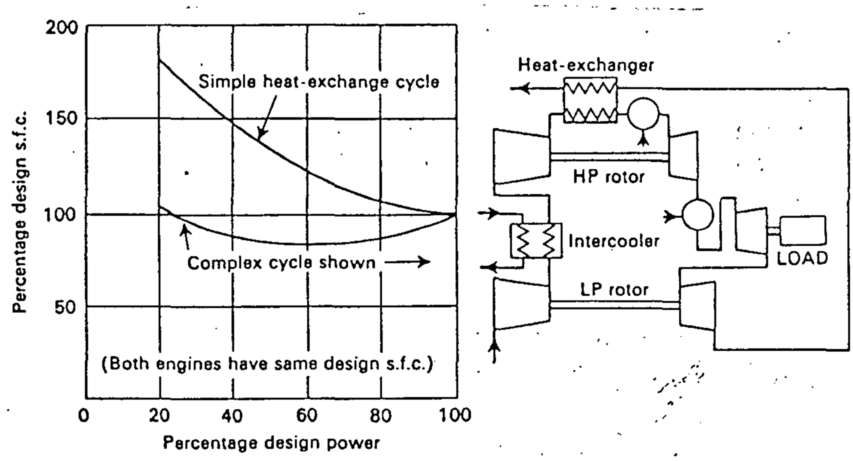


Fig-15.

In practice the turbines are never single stage particularly so incase of land and automotive application. So it is convenient to plot the S.F.C. versus power output seperately for each stage as shown in the Fig-15. When the turbine is driving a load the SFC can be obtained by superimposing all these curves for that particular load. This is shown by the dotted line in the Fig-15. It is clear that the compressor speed is directly dependent upon the power turbine speed. The SFC i.e. shown in the Fig-15 increases the power out increases or vice versa, because the fuel flow decreases as the compressor speed decrease when there is a decrease in the power out put of the turbine. The poor part load economy is a major disadvantage of a simple gas turbine. The part load performance can be improved substantially by use of more complex cycles.

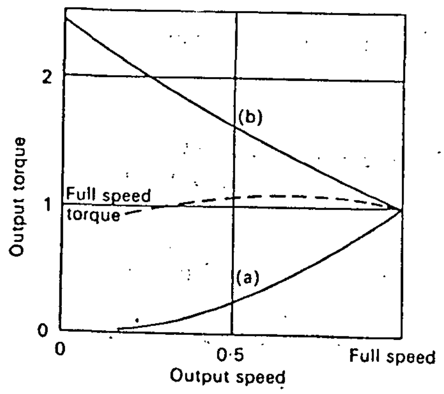
35. For gas turbines where the application is for vehicular or Naval application the SFC has to be imporved because considerable portion of the running time is spent at low power. The methods of improving part load performance is discussed in para 48. The plot of SFC versus popwer out is shown in the diagram Fig-16. Here the plot is for both simple heat exchange cycle and complex cycle :



Part load s.f.c. for simple and complex cycles

Fig-16.

36. <u>Torque charecteristic.</u> As discussed earlier generally the turbines for Land application is of twin shaft type i.e. a separate turbine driving the compressor known as LP/HP turbines and power turbine from which the power output is tapped for coupling to the load. This arrangement ensures the compressor to operate in the optimum efficiency range. The torque is an important parameter which is looked into when selection of any turbine is made. The plot of torque charecteristic is shown in Fig-17.



Torque characteristics

Fig-17.

It is amply clear from the fig that the torque increases as the out speed decreases as the output power remains retatively constant over a wide load speed range for a fixed compressor speed. The maximum torque developed by a gas turbine is as much as three times the torque developed at full speed. The efficiency of the torque conversion at low speeds, for example when accelarating from rest is very low. A gear box is essential to over come this problem which will be generally in the range of four to five speeds incase of heavy vehicles.

ADVANTAGES OF GAS TURBINE OVER IC ENGINES.

37. **<u>Warmup time.</u>** After the engine has been brought upto speed by the starting motor and the fuel ignited, the gas turbine engine will accelarate from a cold start to a full load without any warmup period. This is particularly important for all practical applications like Locomotive, Marine, Aviation etc.

38. **Simplicity.** The construction of Gas Turbine is very simple which consists of compressor connected by a shaft and the gear trains that drive the auxiliaries are the only moving or rotating parts in the system. There are no unbalanced forces so that the engine is vibrationless and the lubrication is also easy and inexpensive. Also the ignition system is required for starting the combustion after which it is self sustaining. The combustion chamber is inexpensive, light weight and small with a high rate of heat release.

39. **Flexibility.** Since different processes within the cycle take part in seperate components (compressor, combustion chamber, turbine) a great variety in the arrengement of the system is possible which can depend upon the duty, performance desired and the space allocation.

40. **Low Weight and Size.** Can also be termed as power to weight ratio. Gas Turbine has lower specific weight (i.e. Kg Weight/HP) and also lower volume space / HP out put than any of the reciprocating engine. This has a cutting edge over IC engine as any saveing in weight reduces the NGP which increases the mobility of a tank.

41. **Independent System.** Open cycle gas turbines except those having intercooler in the system require no cooling water. If a powder charge or other mechanical device is used in place of an electric starter, no electric power or battery is required. The engine is then self contained and independent of outside power or cooling medium.

42. **Fuels.** The combustion chamber can be designed to burn almost any of the hydrocarbon fuels from high octane gasolene down to heavy Diesel oil including solid fuels. Despite careful design some fuels produce too high a carbon or other foreign deposit in the combustion chamber which will certainly reduce thermal efficiency and hence an efficient method of cleaning should be developed to clear the system of such deposists.

43. Combustion and delivery of power is continous there is no peak and fluctuating pressure becaue of this no fly wheel is essential.

44. **Sensitivity.** A simple open cycle gas turbine is sensitive to changes in the component efficiencies, i.e. a reduction in the component efficiency will rapidly lower the thermal efficiency of the cycle. The component efficiencies may also be lowered by dirt or dust being deposited on the compressor blades, by carbon or other foreign deposits from combustion in the combustion chamber turbine and regenerator is used, by reduced speed such as at part load, and by failure to maintain designed efficiencies with continuous operation. The work ratio, which is a measure of the sensitivity, may be improved by the addition of inter cooling and reheating to a cycle.

45. The open cycle gas turbine is also sensitive to changes in the atmospheric temperature as shown in Fig.18 i.e. an increase in atmospheric temperature will lower the thermal efficiency of the engine. This is a disadvantage as far as land and auto applications are concerned particularly to Indian conditions where an automobile has to operate on both extremes i.e. peak winter where the temp goes to sub zero during summers temperatures shoot up beyond 50^0 C. However, it is advantageous to operate a gas turbine during winters where the ambient temperature is low and also in high altitudes.

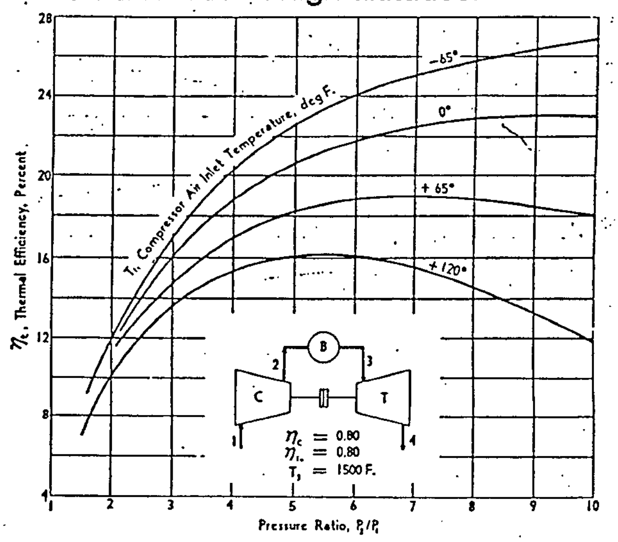


Fig-18.

46. **High air rate.** A simple open cycle gas turbine has a very high air rate as compared to other IC engines. However, this is not disadvantageous in land applications as open cycle gas turbine require higher pressure ratios and higher turbine inlet temperatures for optimum turbine efficiencies.

DISADVANTAGES

47. **Part load performance.** The calculation of the part load characteristics of a gas turbine is very complicated and complex. This system is sensitive to change in component efficiencies, particularly the compressor (axial and centrifugal flow compressors) which have a small speed range of optimum efficiencies. This, coupled with the fact that the air flow rate or load through the compressor cannot be varried without changing the speed which varies the pressure ratio, produces a system that has a low efficiency at part load conditions. The figure-19 below shows the effect of the change in pressure ratio on the efficiency of a simple open cycle gas turbine . This makes it amply clear that gas turbine engines should be run as close to the designed optimum load, usually arround 90-95 percent of full load, as possible.

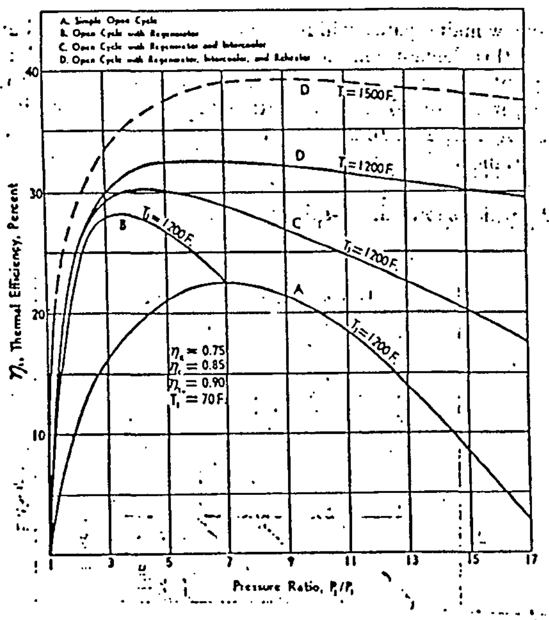


Fig-19.

48. **Improvement of part load performance.** The part load performance of a gas turbine can be improved by various means, i.e. by utilizing inter cooler & reheater in a cycle. Also by using a twin shaft arrangement i.e. one shaft to exclusively drive the compressor at a constant optimum speed so that the inlet temperature is maintained at designed value, part or variable loads at the power turbine are obtained by varying the amount of fuel and therefore the temperature of the working fluid which varries the speed of the turbine. In this manner, part of the cycle is operated at its maximum efficiency while the inefficiency caused by variable speed is contained to only a portion of the system. The effect of the twin shaft arrangement is illustrated by the curves of the Fig-20.

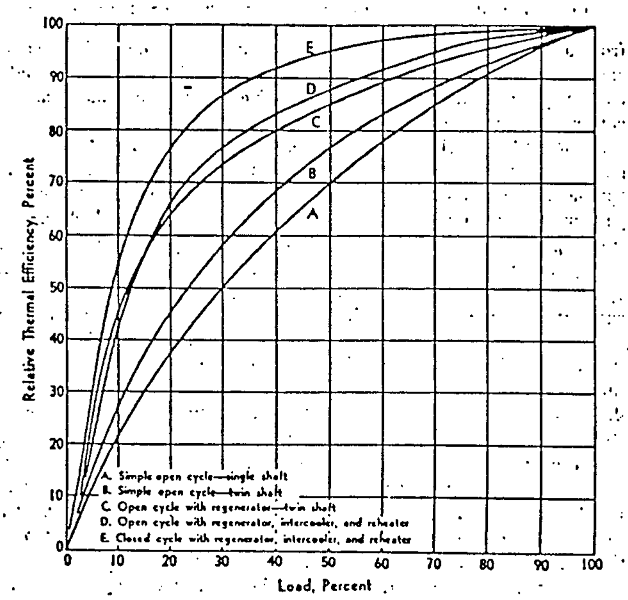


Fig-20.

WHY GAS TURBINES FOR TANKS

50. The mobility of a tank is an important factor on which military planners have been stressing upon out run the Technological developments i.e. taking place in the rest of the world, and scientists all over the world have been striving to improve upon this factor. The land mobility is affected by the various sub

factors as shown in the Fig-21.

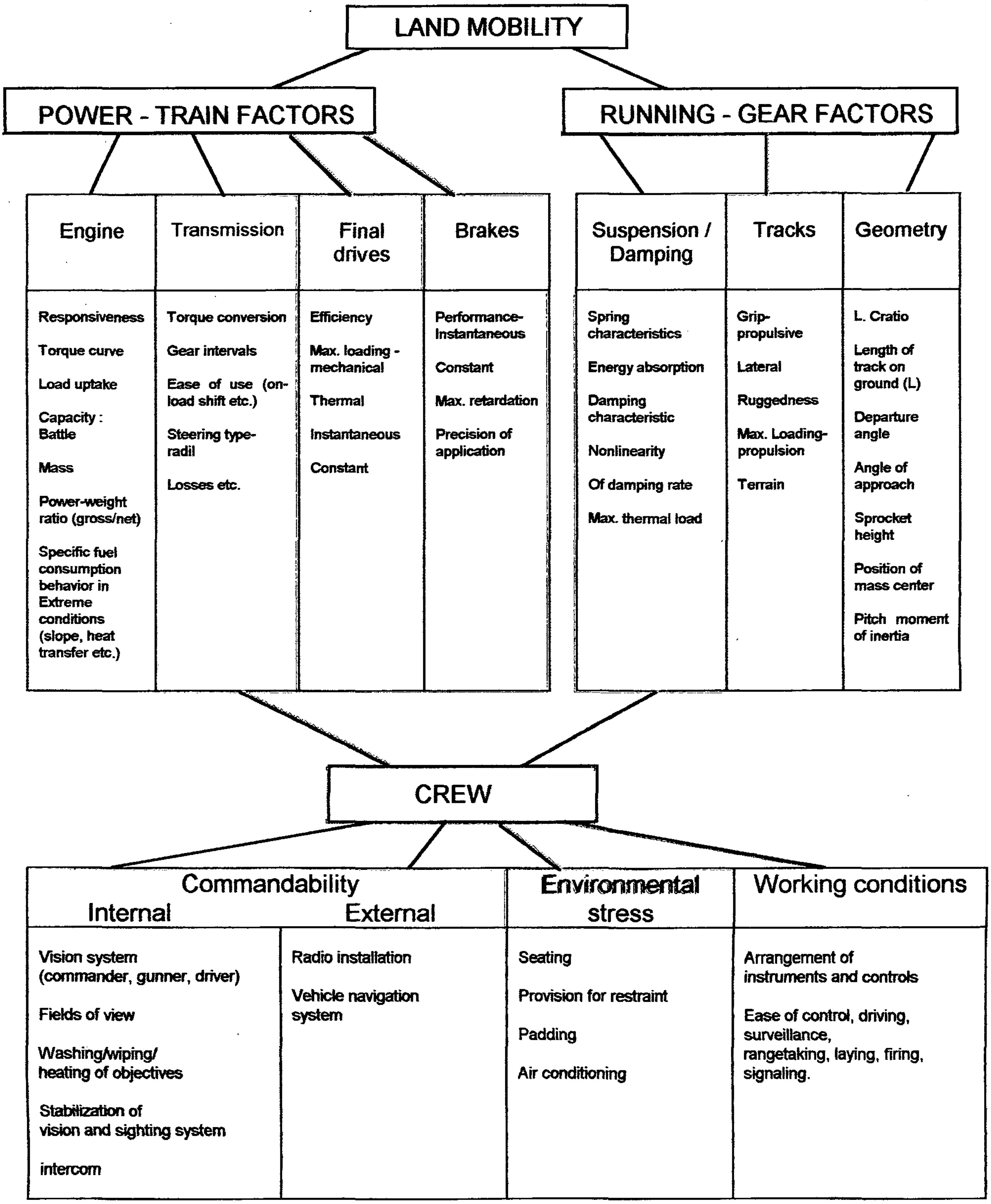


Fig-21.

One of the major assembly that influences the mobility is Engine. More over the volume i.e. occupied by the power packs of T54/55,M-48, Leopard-1, and Leopard-2 are as shown in the Fig-22.

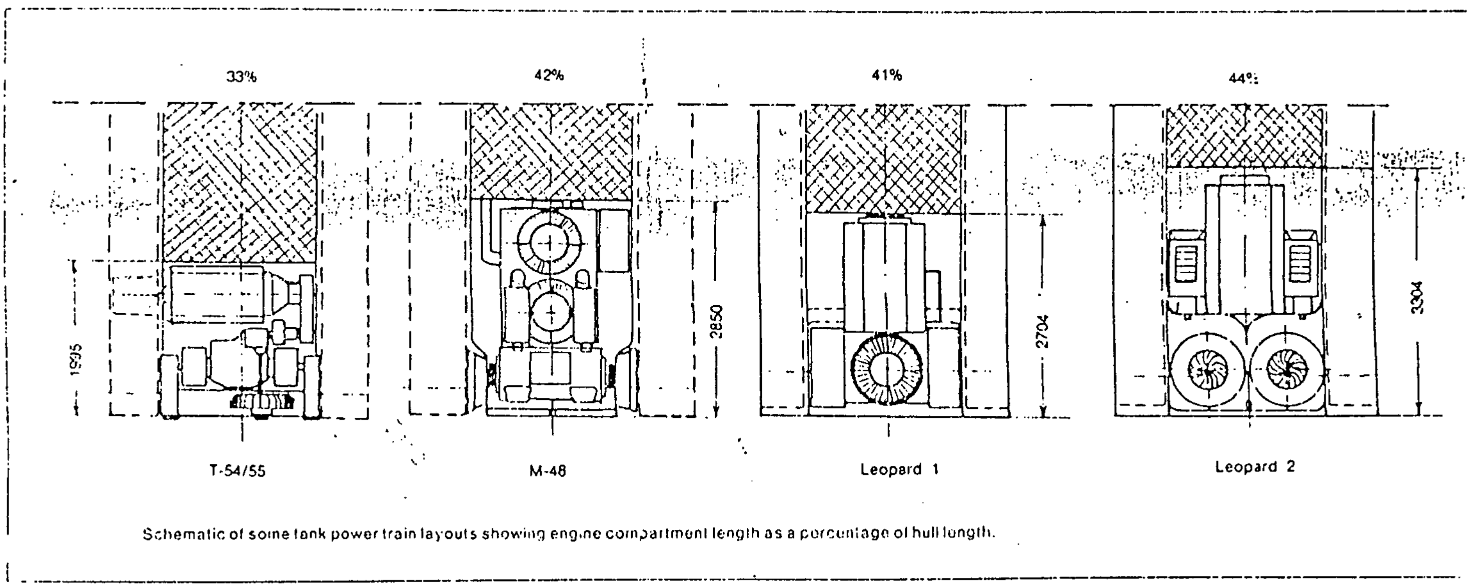


Fig-22.

In the subsequent paragraphs I will enumerate the limitations of diesel engine in reducing the percentage volume that it occupies with that of overhaul volume of AFV.

51. When the parameters specific to the tank application is taken into account, the only way to achieve subtantial improvements in the output of future tank engines is believed to be increasing in the mean effective pressure. Increased capacity would lead to growth in engine size and further increase in speed are limited by problems of development of suitable combustion process and problems of lubrication. The practice in Vogue is, an increase in MEP can only be brought about by supercharging with mechanical blowers or supercharger or exhaust turbo chargers.

52. Going beyond certain boost pressure with exhaust turbo charger adversly affects the ability of the engine to accept sudden loads, and thus the tanks accelaration. It is for this reason that any further increase in the power of

diesel engines for tanks will call for deployment of considerable technical resources. This is clear from the Fig-23. The combustion chamber system, variable control of the exhaust turbo charger, and electronic engine management.

Schematic of key elements of "combustion chamber" system (*left*) and "hyperbar charging" (*right*), both aimed at improving the responsiveness of highly supercharged diesel engines.

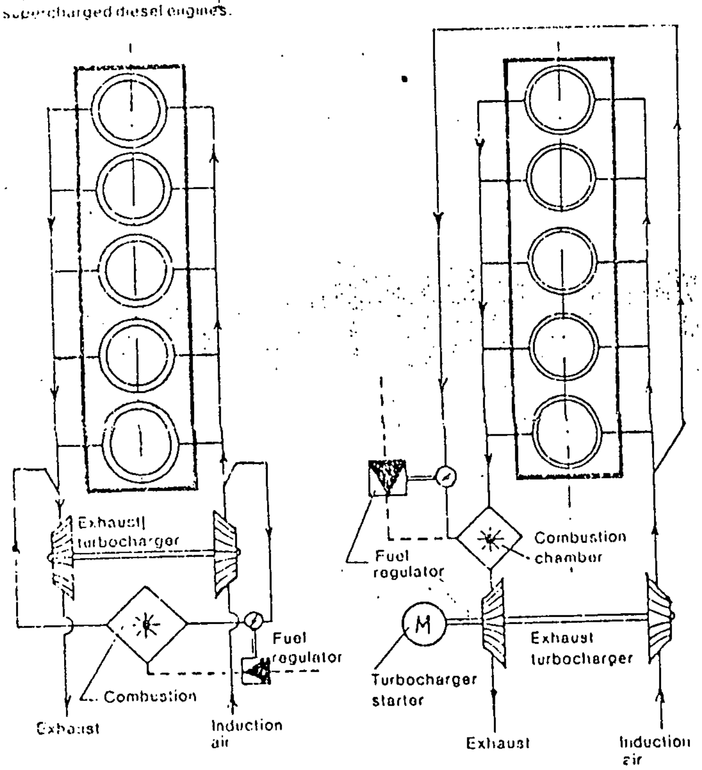


Fig-23.

53. The present efforts demonstrates how, upto the present, all efforts to increase specific output while at the same time reducing capacity have resulted in engines with relatively complex pheripheral subsystems.

54. It is these problems that forces a quest in for an alternative power source for tanks and gas turbine is seen as a potential competitor as against the present diesel engine. Though, not new, the use of gas turbine in tank can be traced back to 1945 with porsche designs for installing a turbo shaft engine in the Tiger-2 Tank. During post war years France has tried Turbomeca gas turbine in 1952 and in mid 50s the United Kingdom carried out an experimental installation in a conqueror hull of parsons type 2979 gas turbine rated at 650 HP and later installation with a 900 HP were trial tested. The advantages as discussed in the later part has made planners to go for gas turbine for AFV applications. The main reason being if the engine occupies increased volume of space in

an AFV that much armour protection has to be provided which will increase the overall weight of the AFV and the net power to weight ratio will decrease. A gas turbine proves to be competitive in solving this problem to quite an extent.

GAS TURBINE DESIGN PROCEDURE

24. Gas turbine design is a complex procedure involving various subgroups like thermodynamics, aerodynamics, mechanical and control system design and emphasizes the need for feed back between the various specialists. The flow chart of design procedure ias as shown in Fig-24.

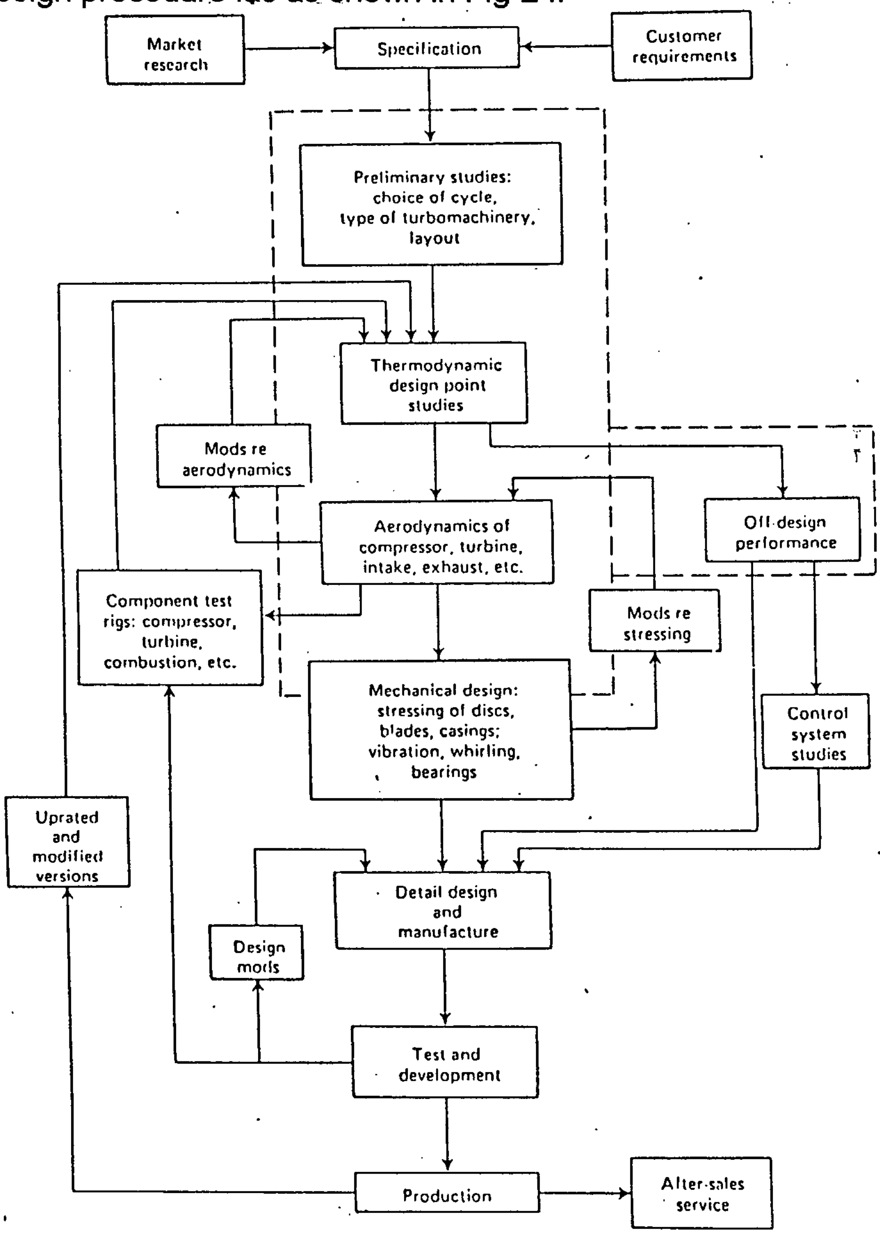


Fig-24.

56. Like any design procedure here also market research is conducted after which specifications is laid down as per the customer requirement. The development of a high performance gas turbine is very costly and generally are developed by multinational consortia. Successful engines are those which have a variety of applications. The specification is just not a simple statement of required power and efficiency. Other factors of major importance , which vary with the application, include weight, cost, volume, life and noise, and many of these criteria act in opposition, for e.g. high efficiency inevitably incurs high capital cost, and a simple engine of lower efficiency may be perfectly acceptable if the running hours are low.

57. The important decision facing the designer is choice of cycle. To begin with it is necessary to decide what type of turbomachinary to use, and this will in large depend on the size of the engine. The layout of the engine must also be considered, for example whether a single or multishaft design should be used.

58. After all of above procedure the first major design step is to carryout thermodynamic design point of studies. These are detailed calculations taking into account all important factors such as expected component efficiencies, air bleeds, variable fluid properties and pressure losses, and would be carried over a reasonably restricted range of pressure ratio and turbine inlet temperature. A value for the specific ouput (i.e. power per unit mass of air flow) and specific fuel consumption will be determined over a range of values of the basic cycle parameters. It should be clearly under stood that there is not a mathematically defined optimum. For example at a given turbine inlet temperature a large increase in pressure ratio may give a minimal improvement in thermal efficiency and the resulting engine too complex and expensive to be pratical. Once the choice of cycle parameter is settled on, the designer can make use of specified output to determine the air flow required to give the specified power.

59. Knowing the airflow, pressure ratio and turbine inlet temperature,attention can be turned onto the aerodynamic design of the turbomachinery. it is now

possible to determine annulus dimensions, rotational speeds and number of stages. At this point a consultation between the aerodynamacist and thermodynamacist interaction is necessary to over come the difficulties arised and if required necessary design changes from thermodynamic point of view is made, perhaps a slight increase in temperature or decrease in pressure ratio.

60. The mechanical design can start only after aerodynamic and thermodynamic design are well advanced,it will then be found that stress or vibration problems may lead to further changes, the requirements of the stress and aerodynamic groups often being in opposition. At the same time as these studies are proceeding ,off design performance and control system design must be considered; off design operation will include the effects of varrying ambient conditions, as well as reduced power operation .When designing a control system to ensure the safe and automatic operation of the engine, it is necessary to be able to predict temperature and to select some of these for use as control parameter.

61. Once the engine has entered service, there will be demands from users for more powerful or more efficient versions, leading to the development of uprated engines. Such demand may offten arise before the design process has been completed. When the engines have to be uprated, the designer must consider such methods as increasing the mass flow, turbine inlet temperature, component efficiencies etc, while maintaining the same basic engine design. A successful engine may have its power tripled during a long development cycle like every system has its peak here gas turbine will also reach its peak, the engine will become out dated and no longer competitive. Again the one more critical point is the decision of timing at which a new design and development should start so that the old set can be replaced in a phased manner.

62. The foregoing gives an overall, if superficial, view of the design process and may lead to the realization that the gas turbine design is a team effort involving a varity of specialist engineers, i.e. thermodynamacist,

aerodynamacist, mechanial and control experts etc.

VOLUMETRIC ANALYSIS

63. It is very important to study the volume i.e. available in the Engine comportment of Tank T-72, mainly because the gasturbine i.e. selected/designed for fitment in the vehicle should confirm to the space available. Any variation of the dimensions on the plus side to that of available space size including that of all accessory drive mechanism will force remodification of the back of the vehicle which will consume additional time and efforts. This will not only take time but also considerable variation in cost as the entire basic change in dimensions will force the production set up to change including change in tooling of the machine tool. The diagramatic representation of space i.e. available is at Fig-25.

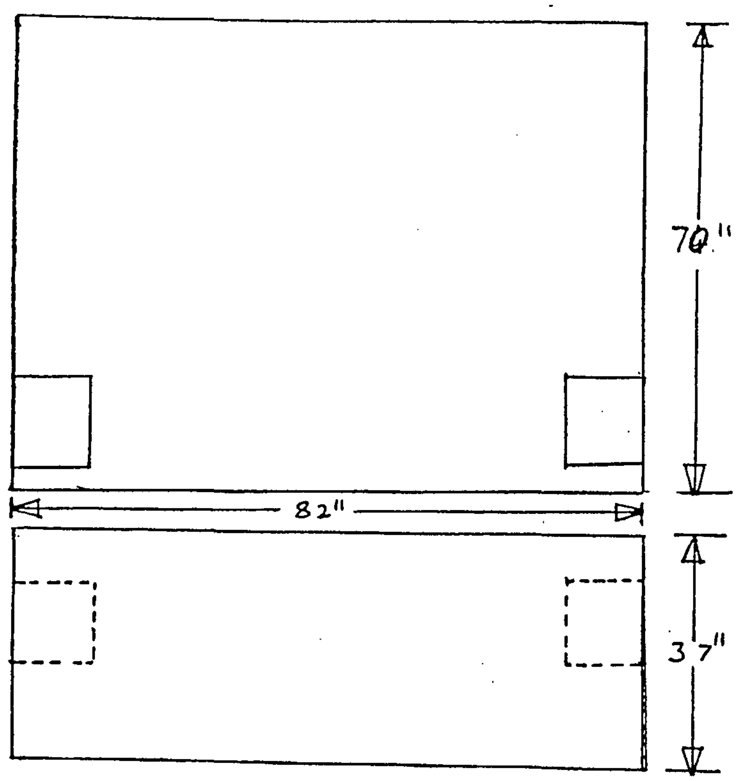


Fig-25

64. The dimenstions that is available width 84" length 70" and depth of 37" with in the space avilable the portion of the Gear box volume has to be excluded. This is the total space available to mount the gas turbine which includes all the accessories i.e. required to drive the gas turbine including that of air filters and reduction gear box that reduces the RPM of the gas turbine from the order of 20000 plus to the operating range as that of diesel engines which is usually between 1500 to 2300 RPM, because at high RPM the torque is very low.

SPECIFICATIONS OF ENGINE V-46-6

65. It is necessory to know the performance specifications of Diesel Engine i.e. fitted presently in Tank T-72. The gas turbine i.e. selected should match the performance specification of the Diesel Engine i.e. the torque and BHP has to be satisfied to maintain the present mobility. The torque and power range may go over the present Diesel Engine design as this will enhance the mobility of the tank. The Transmission need not be redesigned incase the torque and power is more, as some factor of safety would always be incorporated at design stages. Because of this the transmission can take the higher torque and power rating.

(a)	Make	:	V-46-6 multi fuel,12 cycle 4 stroke V-60 direct injection water cooled.
(b)	Horse Power	:	780 at 2300 RPM.
(c)	Max Torque	:	315 $\pm$ 5 Kgm at 1400 RPM.
(d)	Engine capacity	:	38.86 ltrs.
(e)	Specific oil consumption	:	8 g/HP/hr (6 - 11 ltrs in 100 kms)
(f)	Weight of the Engin	:	982 kg.

(g)	Max RPM	:	2300.
(h)	Min RPM	:	800.
(j)	Operating Range	:	1600 RPM to 2300 RPM.
(k)	Specific Fuel consumption	:	180 gm/HP/hr.
(l)	Fuel capacity (with external barrels)	:	1595 ltrs.
(m)	Fuel consumption per 100 km	:	260 to 450 ltrs.
(n)	Cross Country range	:	320 to 480 kms.
(o)	With external barrels	:	420 to 600 km

GAS TURBINES THAT ARE AVAILABLE OFF THE SHELF

66. There are various manufacturers in the world manufacturing a vide range of turbines of various power ratings and applications like Aeronautics, Industrial, Power generation, and Land application etc. Though turbines have numerous advantages over Diesel and Petrol Engines but their applications are very restricted. The manufacturers are, Turbomeca, Pratt & Whitney, Kawasaki, Textron lycoming, Allison, Garret, Solar, Ebara etc. Some of the gas turbines that are manufactured by the aforesaid industries and that meet the specification of the present volume available on the tanks engine compartment are listed from para 71 onwards.

67. More over when evaluating gas turbine design performance it is important to keep in mind that actual out put can differ from nominal performance by as much as 5% depending on how a manufacturer rates the machine, i.e. where he measures the gas turbine output and what he chooses to include in the way of losses.

68. Types of fuel burnt will also makes difference depending upon the manufacturer, the suppliers will quote the variation i.e. better performance on natural gas than on distillate oil from 2% to 3% better while others quote same performance on gas or liquid.

69. Variations in site temperatures and altitude from the design point is most critical. Generally gas turbines are optimized for operation at 15^0 C at sea level conditions. But that can be changed to optimize performance to favour "Off design" conditions imposed by extremely cold or hot ambient temperatures but yet some derating in normal conditions will be imminent.

70. Even if the turbine is selected after considering all the above facts the net out put is determined by installation losses, site conditions, operating conditions as an AFV has to operate at various altitudedes and also wide range of varrying ambient conditions (extremely high summer temperatures of the order of upto 50^0 C in deserts of Rajasthan and extremely low temperatures as low as -30^0 C in Leh region) Fuel used, duty cycle etc. So when ever the selection is made these points are to be kept in mind when selecting a gas turbine.

71. Following are the details of the gas turbines that meet the dimentional requirement of the Engine compartment and brief technial discription specified by the manufacturer.

TURBOMECA BASTAN VI

72. **Description.** Single shaft, axial centrifugal flow, aero derivative, Industrial and marine gas turbine machine can be adopted for land applicaiton with a suitable reduction gear box.

Design Features

73. **In let.** Straight through flow, inlet with four cast support vanes and the inlet is to be modified for vertical flow.

74. **Compressor.** Single axial and single centrifugal flow compressor providing a pressiuve ratio of 5.5:1 at a mass flow of 4.4 kg/sec. The axial stage is made of Aluminium or Magnesium alloy while the centrifugal impeller is made of stainless steel casing or of aluminium and steel.

75. **Application.** Land and Marine Turbines.

76. **Combustor.** Single annular combustor, utilizing torch ignitor for gaseous fuels and spark ignitor for distilate fuels. Composed of outer liner made of multinet and inner liner of Haynes 25. Estimated turbine inlet temperature at 871^{0}C.

77. **Turbine.** A three stage axial flow, uncooled turbine runs at 24,000 RPM. Stage 1-2 blades and discs are made of Nimonic 90 material, stage 1-2 vanes are made of Nimonic 263 and multinet respectively. Stage 3 vanes are made of Nimocast PE10 or IN13LC. Diffuser is made of stainless steel.

78. **Bearings.** The machine is mounted on one roller journal bearing and three ball thrust bearings.

79. **Accessories.** The machine is supplied with hydromechanical fuel control and 24 V electric starter or a compressed air / gas expansion starting system.

80. **Power.** 764 - 858.5 SHP, the variation is because of variations in ambient conditions as discussed earlier.

81. **Dimensions.**

(a)	Length	:	62.99"
(b)	Width	:	27.95"
(c)	Height	:	33.07"

82.	Weight.	:	320 Kg.
83.	Heat Rate	:	17,360 KJ/KWH.
84.	Exhaust Temperature	:	500^{0} C.

85. **Applications.** Industrial, Power Generation, Land and Marine application.

86. **Cost .** : $ 200,000

TURBO MECA BASTAN VII.

87. Most of the features of turbo meca Baston VII is same as Bastan VI except for few variations which are :

88. **Compressor.** It has two axial flow and one centrifugal flow compressor providing increasing engine pressure ratio of 7.2:1 and air flow of 6 kg/sec. As the pressure ratio and air flow rate is more, power generated is more.

89.	Power.	:	1099 BHP.
90.	Dimensions.		
	(a) Length	:	66.93".
	(b) Width	:	27.95".
	(c) Height	:	33.07".
91.	Weight.	:	350 kg.
92.	Heat Rate	:	15,490 KJ/KW.
93.	Exhaust Temp	:	450^0 C
94.	Application	:	Industrial, Power Generation, Land and Marine application.
95.	Cost	:	$ 280,000.

TURBOMECA MEKILA TI

96. **Intake**. Annular pitot type with bullet dome spinner with hot air anti-icing.

97. **Compressor.** Three stage axial compressor and single stage centrifugal unit providing a pressure ratio of 9.9:1 and air flow rate of 5.4 Kg/sec.

98. **Combustor.** Single annular combustor and fuel injected into the combustor via a single centrifugal injector located in the shaft with smiths ignitors.

99. **Compressor Turbine.** Two stage gas producer / compressor turbine drives the gas generator at approx 33,350 RPM with turbine inlet teperature of approximately 1204^{0}C. The gas producer turbine stage 1 - 2 blades are cast in MAR - M - 002.

100. **Power Turbine.** Two stage power turbine drives the rear mounted output shaft at 22,850 RPM, stage 1 & 2 blades are made of MAR-M-004 (IN 713.LC+Hi) power take off takes at 14,000 - 23,000 RPM.

101. **Accessories.** Top mounted accessories gear box driven via a shaft from the compressor rotor, with electric or pneumatic starting. Full authority digital electronic control and turbo meca fuel control. For industrial application reduction gear boxes available at 5000, 1800 & 1500 RPM.

102. Power	:	1475.5 SHP base ratting . 1609.5 SHP peak rating.
103. Dimensions		
(a) Length	:	87.8"
(b) Width	:	27.6"
104. Weight dry	:	350 Kg.
105. Specific weight	:	0.3 Kg/HP.

106. Heat rating : 12,918 KJ/KW.

107. Efficiency : 28%

108. SFC : 213 g/HP-hr.

109. Exhaust gas temperature : 505^{0}C.

TURBOMECA INDUSTRIAL TURMO 111

110. **Inlet.** Annular intake with bullet dome housing covering accessory gearing for below inlet mounted accessory drives.

111. **Compressor.** Single stage axial and single stage centrifugal compressor producing a power ratio of 4.9:1 with a mass flow rate of 5.4 Kg/sec. The compressor runs at 31,100 RPM.

112. **Combustor.** A single annular combustion chamber in corporating dual torch or spark igniters and dual fuel nozzles fed by standard centrifugal injection wheel. Turbine inlet temperature is estimated at 899^{0}C.

113. **Gas Generator Turbine.** A dual axial-stage turbine drives the gas generator at its rated speed. Both stages are uncooled and made of udimet - 700 blades and discs. Stage 1 vanes are made of Nimonic 75 & Hayes 25 while Stage 2 Vanes are made of Nimocast PE 10.

114. **Power Turbine.** Two-stage, axial flow turbine rotate at a maximum speed of 15,850 RPM and provides a 14,500 - 20,500 RPM range of operation with single or two-stage gearing provides O/P speeds of 5,500 RPM and 1500 - 6000 RPM respectively. Power turbine discs are made of Nimonic 90, while all blades

are either Nimonic 90 or IN 713. Stage 1 vanes are made of Nimonic 75 while stage 2 vanes are cast in Nimocast PE 10.

115. **Bearings.** Gas generator is carried on the roller journal bearing and single ball journal bearing while the free power turbine and O/P drive shafts are carried on a single ball and single roller bearing and twin shell bearings respectively.

116. **Accessories.** Utilizes hydromechanical control system, with 24 V DC starter (Optional). Fuel injection pump utilizes a eight stage PVMX Centrifugal pump.

117.	Power	:	1005. 5 SHP base power.
		:	1207. 5 SHP peak power.
118.	**Dimensions.**		
	(a) Length	:	82.67".
	(b) Width	:	29.92".
	(c) Height	:	28.74".
119.	Weight dry	:	355 Kg.
120.	SP Weight	:	0.353 Kg/Hp.
121.	Heat Rate	:	18,750 Kg/Sec.
122.	Exhaust Gas Temperature	:	510^{0}C
123.	Cost	:	$ 250,000

TURBOMECA TURMO XII

124. This is an improved version of TURMO III. The improvements that have been made are:

125. ***Compressor.*** Dual axial stage compressor followed by a single centrifugal type compressor with pressure ratio of 8.2:1 and mass flow rate of 6.94 Kg/Sec.

126.	Power	:	1341.5 HP Base power.
		:	1609.5 HP Peak power.
127.	Dimensions		
	(a) Length	:	82.67"
	(b) Width	:	27.95"
	(c) Height	:	34.25"
128.	Weight	:	490 Kg.
129.	SP Weight	:	0.37 Kg/HP
130.	Heat Rate	:	15,520 KJ/KW
131.	Mass flow rate	:	6.94
132.	Exhaust gas temperature	:	450° C

133. Cost : $ 360,000

TURBOMECA MAKILA TI
(vehicular/rail propulsion)

134. **Intake.** Annular pitot type with bullet dome spinner and hot air anticing.

135. **Compressor.** Three stage axial flow compressor and single stage centrifugal unit, with pressure ratio of 9.9:1and mass flow rate of 5.5 Kg/sec.

136. **Combustor.** A single annular combustor and Fuel is inject by single centrifugal injector located in the shaft

137. **Compressor Turbine.** Two-stage gas producer / compressor turbine drives the gas generator approximately at 33,350 RPM. Turbine inlet temperature approximately 1204^0 C.

138. **Power turbine**. A two stage PT drives the rear mounted output shaft at 22,850 RPM. Two nozzles are produced in HS 31 (x-40) cobalt-base alloy and stage 1-2 blades in MAR-M004 power take-off at 14,000 - 23,000 RPM. Exhaust gas temperature is estimated at 525° C.

139. Power	:	1609.5 SHP Base power.
	:	1676.5 SHP Peak power.
140. Reduction Gear box	:	Available at RPM ranges of 5,000, 1800 & 1500

141. Dimensions

(a)	Length	:	87.8"
(b)	Width	:	27.6"
(c)	Weight dry	:	440 Kg.

142.	Heat rate	:	12860 KJ/Kwh
143.	SFC	:	213 g/HPm
144.	Cost	:	$ 390,000

TEXTRON LYCOMING AGT 1500

145. This engine was developed exclusively for Tank XM1 Abrams sponsered by US Department of Defence. This tank is in service with American Army and Marines. Also exported to Saudi Arabia, Egypt and also Egypt has been granted licence to produce these tanks.

Design Features.

146. **Intake.** Air enters vertically through a vehicle mounted two stage air filter, with internal particle seperator and a barrier filter which is turned 90° to flow axially to compressor.

147. **Low pressure compressor.** Five stage, axial flow stainless steel low pressure compressor of constant outside diameter, with variable inlet guide vanes. The compressor is mounted in two piece compressor case.

148. **High pressure compressor.** HP compressor consists of four-stage axial-flow unit followed by a single centrifugal compressor. The HP counter-rotates relative to the LP compressor. A pressure ratio of 14.0 - 14.5 is maintained. The axial stages are of stainless steel, centrifugal unit of steel.

149. **Recuperator.** The high pressure air from centrifugal stage travels radially and turns 90° to flow through the internal passages of the recuperator to gain some of the heat from the exhaust and enters combustion chamber.

150. **Combustor.** Single tangential scroll type reverse flow annular combustor, with single spark igniter and fuel nozzle. Combustion air enters at 566° C at 1420.3 Kpa.

151. **High pressure turbine.** A single, aircooled HP axial turbine drives the high pressure compressor. Disc is forged waspaloy and turbine blades are air cooled with turbine inlet temperature of 1193° C.

152. **Low Pressure Turbine.** A single uncooled axial stage drives the LP compressor via the inner coaxial shaft.

153. **Power turbine.** Two-stage uncooled axial flow turbine that drives the O/P shaft and a reduction gear box reduces the 22,500 RPM to 3000 RPM.

154. **Accessories.** The auxillary gear box is located beneath the compressor housing and is driven from the compressor shaft. A vehicle hydraulic pump is driven by the gear box which provides the hydraulic pressure to the vehicle systems.

155. Power	:	1500 SHP - AGT 1500
		1500 SHP - AGT 1500 TME
		1675 SHP - AGT 1500

156.	Torque	:	380 Kg - m at 3000 RPM output speed.
157.	SFC (AGT 1500)	:	224 gm/ HP/Hr at Full power. 215 gm/ HP/Hr at 1200 SHP. 203 gm/ HP/Hr at 1500 SHP (with Digital Electronic Control Unit)
158.	Dimensions		
	(a) Length	:	66.63"
	(b) Width	:	8.93"
	(c) Height	:	33.46"
159.	Weight (dry)	:	1134 Kg
160.	SP Weight	:	0.67 Kg/5 HP
161.	Cost	:	$ 315,000 For AGT 1500 TME propultion system.
162.	Road Speed	:	75 KM / hour.
163.	Fuel Capacity	:	1900 Ltrs.

164. Range : 450 Kms (without refueling).

(Travelling at a speed of 40 KM / hour with continous operation in 24 hrs combat day)

165. Gradient : 32°

166. Vetricle obstacle : (1.25 m)

167. Trench : 2.75 meters.

168. Engine life : 20,000 kms without any major overhauls.

169. Power to weight ration : 27 HP/T

170. Accelaration : 0-32 Kms in 6.85 secs.

171. Approximately 66% maintenance can be carried out without removal of Engine.

172. If removal of engine is necessory it takes only 25 minutes.

173. The enhanced parameters like speed and mobility made U.S. Army to develope new tactics.

T - 80

174. Tank of eighties developed by earstwhile USSR, presently the tank is being manufactured at Ukraine also. Few of them have been exported to Pakistan and licenced production negotiations are under progress. This tank is development of T-69. The same transmission system is also used. Though not much of data could be obtained whatever is available is listed below.

175.	Power	:	1250 HP
176.	Road speed	:	75 KMPH
177.	Range	:	400 KMS
178.	Verticle obstacle	:	0.9 m
179.	Gradability	:	60%
180.	Trench	:	2.7 m
181.	Combat weight	:	43,000 Kg
182.	Power to weight ratio	:	22.9 HP / Ton

183. The program of continued fitment of gas turbine was disbanded because it was forseen that the gasturbine will incur additional administrative burden.

184. Other engines that are available in the world that fits into the space available along with reduction gear box and air cleaner.

TURBINE MODEL	STLL-795	ST6L-813	ST6T-76
COMPANY	PRATT AND WHITNEY		
POWER RATING IN SHP	862	1048	1447
MASS FLOW IN KG/SEC.	3.5	3.8	5.86
TURBINE SPEED	33000	30000	6600
EGT	600° C	578° C	575° C
WEIGHT	306 LB	360 LB	740 LB
LENGTH	48"	48"	72"
WIDTH	24"	24"	48"
HEIGHT	24"	24"	36"
TORQUE AT 2000 RPM	308 KGM	374 KGM	345 KGM

TURBINE MODEL	IM 150-1988	IM 150-1988
COMPANY	ISHI KAWAJIMA - HARIMA HEAVY INDUSTRIES	
POWER RATING	1508 SHP	1508 SHP
PRESSURE RATIO	9.9	9.4
MASS FLOW	5.36 KGS	5.36 KGS
TURBINE SPEED	22678	22746
EGT	890° C	890° C
WEIGHT	440 KG	440 KGS
LENGTH	67"	67"
WIDTH	24"	24"
HEIGHT	24"	24"
TORQUE APPROX.	380 KGM	380 KGM

185. XMI by US and T 80 by Russians are not just the AFVs where the gas turbines have been tried out. But also there are many other AFVs, Artry SP guns, Cars and Commercial vehicles the gas turbines have been fitted and is in service in many countries. In vehicular gas turbine market the Garrett GT 601 twin speed

free turbine recuperated open cycle continues to be the best automobile derivative in terms of power output and fuel economy. Development of 750 SHP gas turbine was begun by Industrial Turbine International comprised of Garret, Mach Trucks and the KHD, under the Tacon Contract. The GT 601 had been tested in XM 723 vehicle, and by FRG Army in a U.S. M-48 Tanks.

186. The other military vehicles in which the gas turbine engine has been tested, are :

(a) FRG in Marder

(b) UK in Chieftain (in 1984)

(c) USA BMY/Army M69 155 mm SP Howitzer (in 1985)

(d) France GIAT AMX 30 MBT

(e) Israel T-Tank

187. **Turbomeca TM 403.** The TM 403 rated at 450 - 600 SHP is a vehicular derivative of Arriel aviation turbo shaft engine was initially used in Thomson - CSF / Panhard P6R six wheeled vehicle offered as radar aquisition unit and missile launching unit for the Shahine / SICA anti aircraft missile system. Here the turbine TM 403 drives an alternator of 110 KVA power from were the power is diverted to six traction drive motors, one per wheel.

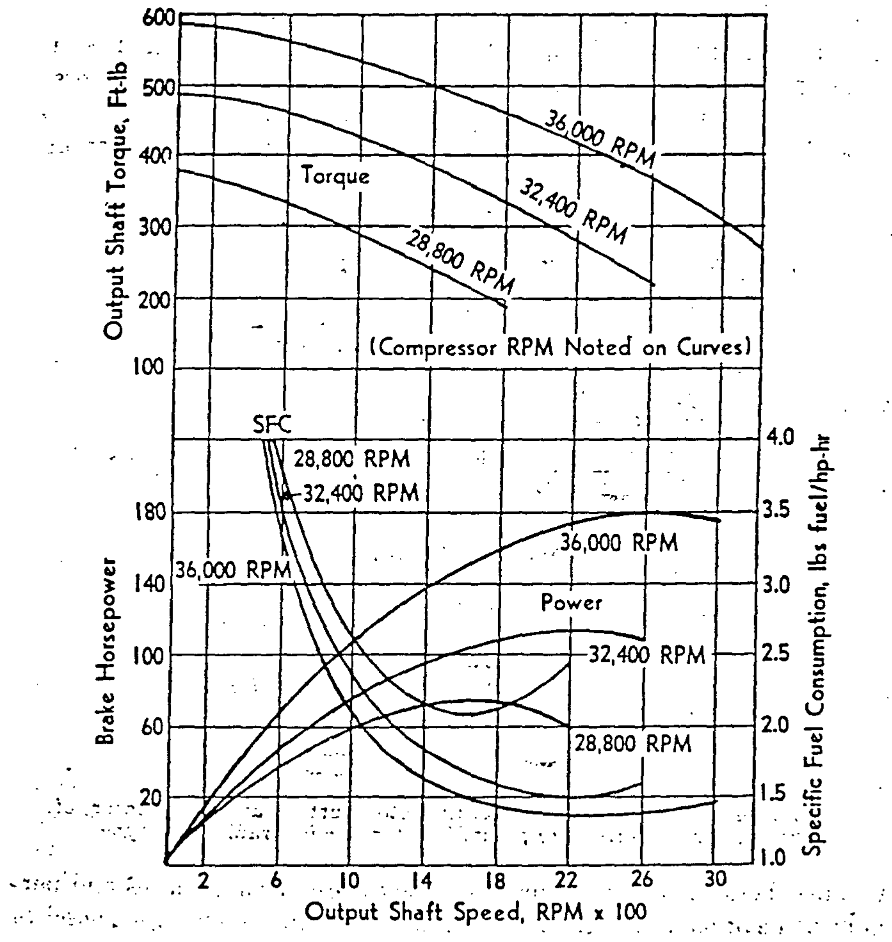
ANALYSIS

COMPARITIVE TABLE OF ENGINES UNDER STUDY

ENGINE MODEL	V-46-6 T-172	AGT 1500	BASTAN VI	BASTAN VII	M-TI	T-III	T-XII	M-TI R/L
POWER HP	780	1500	858	1099	1609	1207	1609	1676
TORQUE AT	315 AT 1400	380 AT 3000	307 AT 2000	46.9 AT 16754	385 AT 3000	346 AT 2500	385 AT 3000	400 AT 3000
WEIGHT IN KGS	980	1134	320	350	350	355	490	440
SFC G/HP/HR AT FUEL POWER	180	WITH DECU 204.4	--	--	213	--	--	213
LENGTH	82"	66.63"	62.99"	66.93"	87.8"	82.7"	82.7"	87.8"
WIDTH	72"	8.92"	27.95"	27.9"	27.6"	29.9"	27.9"	27.6"
HEIGHT	37"	33.46"	33.07	33.07"	--	28.7"	34.3"	--
COST $ IN THOUSANDS	N K	315	200	280	250	250	360	390
WETHER FITS IN THE AVAILABLE SPACE	YES	YES	YES	YES	NO	WITH MINOR MOD	WITH MINOR MOD	NO
POWER TO WEIGHT RATIO	18.57	35.7	20.4	31.6	37.4	28	37.4	38.97

SELECTION OF A SUITABLE GAS TURBINE

188. Having seen the gas turbines that are available in the world it is essential to select a turbine that is superior to the present diesel engine so that an improved mobility feature is achieved and adequeate room for enhancing other charecteristics i.e. protection can be posible.



600
500
400
300
200
100
Output Shaft Torque, Ft-lb
36,000 RPM
32,400 RPM
28,800 RPM
Torque
(Compressor RPM Noted on Curves)
SFC
28,800 RPM
32,400 RPM
36,000 RPM
36,000 RPM
Power
32,400 RPM
28,800 RPM
180
140
100
60
20
Brake Horsepower
4.0
3.5
3.0
2.5
2.0
1.5
1.0
Specific Fuel Consumption, lbs fuel/hp-hr
2
6
10
14
18
22
26
30
Output Shaft Speed, RPM x 100

FIG-26

189. Sufficient efforts are on to uprate the present engine to atleast 1000 HP, already uprating upto 1080 has been achieved at CVRDE and is presently under fitment by way of removing the supercharger and incorporating the turbo charger. Hence Bastan VI of 858 HP is left out. Other engines like Mekila TI and Mekila TI Rail and Land application is not selected as the size of the engine is bigger than the space available and if selected for fitment a major modification to the hull is essential which will be a costly affair. Turmo III and Turmo XII is also left out because only the engine can be accomodated and there will be no space for reduction gear box and air cleaner and the engine is more powerfull too which demands a change of entire transmission system.

190. Though AGT 1500 can be accomodated in the available space with air cleaner and reduction gear box the enigne of 1500 HP which is generally required for 50 ton plus class of A vehicles. The cost of the gas turbine is quite exorbitant and hence not recommended.

191. Bastan VII can be easily accommodated in the available space along with reduction gear box and aircleaner. The power of the turbine is 1099 that gives a power to weight ratio of approximately 26 HP/Ton and is quite a satisfactory improvement, also the cost of the turbine is about $ 250,000/-. The torque developed is 47 Kgm at 16754 RPM which can be increased upto 450 Kgm with suitable reduction gear box. With a fairly good specific fuel consumption of 213 g/HP/Hr proves to be fairly economical as compared to that of other gas turbines. Hence Bastan VII is selected.

AIR CLEANER

192. One of the main requirement of any engine is an efficient aircleaner. This is mainly because the dirty air entering the system will enhance premature wear as the dust will act as grit between the working parts. Incase of gas turbine the uncleaned air will not only enhance the wear but also drastically affect the

component efficiencies i.e.the efficiency of compressor, combustion chamber, turbine etc. The change in efficiency of compressor is due to deposit of dust on the compressor blades which will reducde the RPM of the compressor and hence air flow rate is affected. In view of the afore said reasons that an efficient air cleaning system is essential weather it is diesel engine or gas turbine. A gas turbine offers flexibility of accepting a bit bigger grain size of dust as compared to that of diesel engine. Here I am going to discuss the requirements and design of air cleaner for gas turbine.

REQUIREMENTS OF AIR CLEANER

193. System used to filter the induction air of tank engines are subjected to a special set of parameters. The requirements are

(a) The system should be able to filter out dust of grain size upto 6-7 microns.

(b) The air rate i.e. required for a gas turbine is as much as three times as compared to that of diesel engine. So the system should be able to supply the required quantity of air for protracted periods under adverse dusty conditions while maintaining the efficiency.

(c) Effective filteration must be maintained at extreme angles i.e. on slopes of upto 60% ($55\ ^0$).

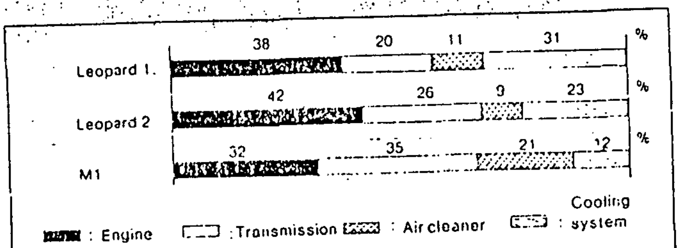
(d) Intervals between cleaning must be as long as possible under extreme conditions of dust, upto $3g/m^3$. As far as possible the system should be self cleaning type.

(e) The filters should be as resistant as possible to incendiary agents such as NAPLAM.

(f) If the air cleaner is not efficient the exhaust temperature will be more, burnout the seals enroute to exhaust, the boost pressure will drop etc.

(g) A fan required to be provided to extract the dust instead of present system of dust extraction that is in vogue incase of T series engine to avoid failure & extra ducting i.e. required. The dust extracted incase of Leopard 1, 2 and M 1 per 100 km is shown in Fig-27.

The position of the engine air intake is critical for the dust loading of the air cleaner. Note (*arrowed*) the cunning arrangement on LEOPARD 1. Subsequently the fitting of skirt plates reduced the dust loading further.



Volume analysis of elements of tank power trains.

Amounts of dust to be removed by air cleaners per 100 km run, for various tanks.

Conditions: Dust content			3 gm/m³
Average speed			35 kph
Average engine load			50%
	Leopard 1	Leopard 2	M 1
Air throughout at nominal load (m³)			
per minute	52.2	105	285
per hour	3132.0	6300	17100
Dust extracted per 100 km under above conditions (kg)	13.4	27.0	73.3

Fig-27.

199. <u>Air Cleaner Design.</u> Though various combinations of air cleaner is possible i.e. uniflow cyclone followed by silk/paper filter stage or reverse flow cyclone followed by steel wool soaked in oil. Here design is dealt with the combination uniflow followed by paper filter element.

(a) Dust concentration for heavy vehicle application is 1600 mg/cubic metere in zero visibility conditions. But the filteration should be efficient upto two times the concentration at zero visibility conditions.

(b) An uniflow cyclone can filter dust size upto 25-30 microns and the efficiency of 94-95%. An uniflow cyclone has a pressure drop of 35 cms of water from inlet to out let where as a reverseflo cyclone has a pressure drop of double this. Hence if a reverse flow cyclone is selected the size of the air cleaner will be very big hence uniflow cyclone is selected for this purpose.

(c) The second stage paper filter size if os paper filter of sieve size 7 microns so that dust size upto 8 microns will be filtered out and further the filtering efficiency will increased due to reduction in sieve size because of continous working and change of filter element will be necessary. Otherwise the quantity of air that is supplied will reduce and hence the power drop.

(d) Following are the some of the extracts of BS1707 & SAE 737 hand books which gives the details i.e. required for designing an air cleaner.

(i) Dust concentration of various types of vehicles.

(aa) Light duty 200 mg/cubic meter.

(ab) Medium 500 mg/cubic meter.

(ac) Heavy duty 1600 mg/cubic meter.

(ii) Details of cyclone.

(aa) Standard size of cyclone (diameter) 23 mm

(ab) Pitch 25 mm

(ac) One cyclone can provide 0.48 cubic meter/min.

(ad) Height of the cyclone 90 mm.

(e) Calculation of no of cyclones required.

Required air rate of turbine = 6 kg/sec.

= 360 kg/min.

15% additional air to cater fo air extracted along with dust particles and the pressure drop.

The air rate for air cleaner calculation

$$= 360 + \frac{360 \text{ X } 15}{100}$$

= 414 kg/min.

Density = mass/vol = kg/cubic metere

= 414 kg X cubic metere/kg/min

The required air rate = 414 cubi meter/min.

No of cyclones required = 414/0.48

= 862.5

= 860 cyclones

(f) Area of paper required for air rate of 414 cubic meter/min.

Specific flow rate of paper/cloth = 500 sq. Meter/c mtr/min.

Area of paper required for a flow rate of 414 c mtr/min.

$$= \frac{414 \times 500}{100 \times 100} \times \frac{c\ mtr}{min} \times \frac{sq.\ mtr}{c\ mtr} \times min$$

$$= 207 \text{ sq. Mtr}$$

(g) Recommended profile of air cleaner is shown at Fig-28.

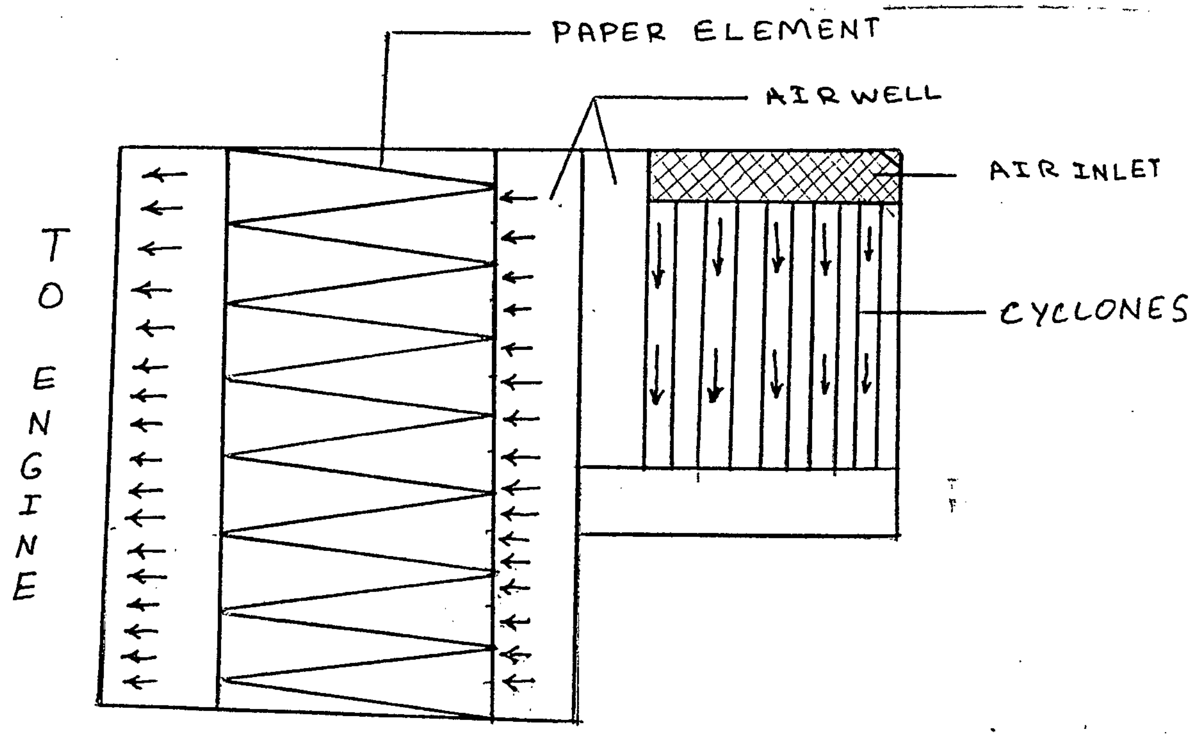


Fig-28.

(h) Size of the air cleaner since the requirement of no of cyclones and the area of paper element is known the profile of the air cleaner can be modified to the space i.e. available inside the engine compartment and ducting.

(j) Dust loading to air cleaner can be reduced to considerable extent by providing skirting plates louvers mesh etc.

195. <u>Tests of Air Cleaner.</u> Dust holding capability of air cleaner can be determined by

$$W = k \times v \times n \times d$$

Where

K = is a factor on the type of engine i.e. whether gas turbine/two stroke/four stroke engine and is

2.352	for gas turbines
1.176	for two stroke engines.
0.588	for four stroke engines.

v	=	volume of air required.
n	=	governed engine speed.
d	=	dust/air ratio selected refer para (ii)

efficiency of the air cleaner

$$\text{efficiency} = 100\,\frac{C-D}{C}$$

c	=	weight of dust fed into air cleaner (known)
d	=	weight of dust transmitted by the air cleaner.

197. The above checks can only be found out by practically subjecting the air cleaner built for any specific operation and is required to be modified incase donot meet the specifications.

REDUCTION GEAR BOX

198. The gas turbine operates at very high RPM normally in the order of 20,000 plus. At this speed the torque is very less and the RPM is not practicable to be used for tank applications. The RPM has to be brought down to the range of 1500 to 2500 and at this range of speed the torque i.e. developed will be of some

use in heavy vehicle application. Hence a reduction gear box is required, summarising the requirements of reduction gear box.

(a) To bring the RPM of the selected gas turbine from 16754 RPM to 1800 RPM.

(b) To multiply the torque of 47 kgm developed at 16754 to that of around 400 kgm so that the initial starting torque and subsequently the high torque requirement to negotiate obstacles and cross country movement demand can be met comfortably.

(c) To act as a link between gas turbine and main gear box so that the motion can be transmitted to gear box from gas turbine.

199. Though it was not possible to locate a reduction gear unit which can fit into the space available I have designed a gear box which can fit into the space available inside the engine compartment in place of transmission gear unit keeping in mind the available distance between gas turbine output shaft and gear box shaft. The constraints under which the reduction gear box that has been designed are.

(a) The centre to centre distance i.e. available is 784 mm.

(b) The present gear ratio of 1.53846 i.e. the ratio of RPM at max power (2000 RPM) to that of RPM at maximum torque (1300 RPM) is maintained as I donot intend changing the side gear box i.e. already existing on the tank. The side gear box i.e. fitted in TK T-72 has been used in TK T-80 U without any change where the power of gas turbine is 1250 HP. Hence the gear box can sustain the power of the present gas turbine (1099 HP) without redesign for the strength and design of gears for transmission for constant power and constant torque. This point was confirmed by CVRDE from the manufacturer in Ukraine through a letter. Presently the T-72 engine has been uprated with turbo charger to 1080 HP. The air cleaner and the cooling system has been redesigned keeping the basic dimensions of engine and transmission same. The torque developed by the uprated engine is more than 400 kgm.

(c) The power of the turbine is 1099 HP at 24035 RPM. Unlike diesel engine the torque characteristic of a gas turbine is regenerative i.e. the torque increases as the RPM reduces and the stall torque goes upto as much as three times the torque developed at maximum power. Hence the RPM at maximum torque i.e. essential in designing the gear box can conveniently selected on the torque curve. Here it is 47 kgm at 16754 RPM. So the reduction gear box is designed to operate between 1800 - 2600 range at output stage with a reduction ratio of 9:1.

(d)The cooling requirement of gas turbine is one fourth that of diesel engine as the cooling system is requir(ed to cool only the engine oil and the heat carriedaway by the gear oil due to heat generated by transmission system.

(e) The gear ratio of present gear box is 1.5384 and the calculated gear ratio for gas turbine is 1.44. The difference between the gear ratios is minimal and is assumed that the gear box will perform sastisfactorily with the selected gas turbine also.

200. Design Procedure.

(a) The gear ratios of present side gear box are forward.

I	Gear	8.173
II	Gear	4.4
III	Gear	3.485

IV	Gear	2.787
V	Gear	2.021
VI	Gear	1.461
VII	Gear	1.0
Reverse Gear		14.35

(b) T-72 engine develops 780 HP at 2000 RPM and 315 $\pm$ 5kgmat 1300RPM.

$\therefore$ Gear ratio = 2000/1300

= 1.53846

(b) Gas turbine

1099 HP at 24035 RPM

47 kgm at 16754 RPM

With a reduction ratio of 9:1 the maximum / min. RPM will be reduced to 2600 to 1800 RPM.

$$\text{Gear ratio} = \frac{2600}{1800}$$

= 1.44

Reduction ratio require = $\frac{16,754}{1800}$

$= 9.3025$

(c) Mechanical schematic of the reduction gear box is as shown in the Fig-29.

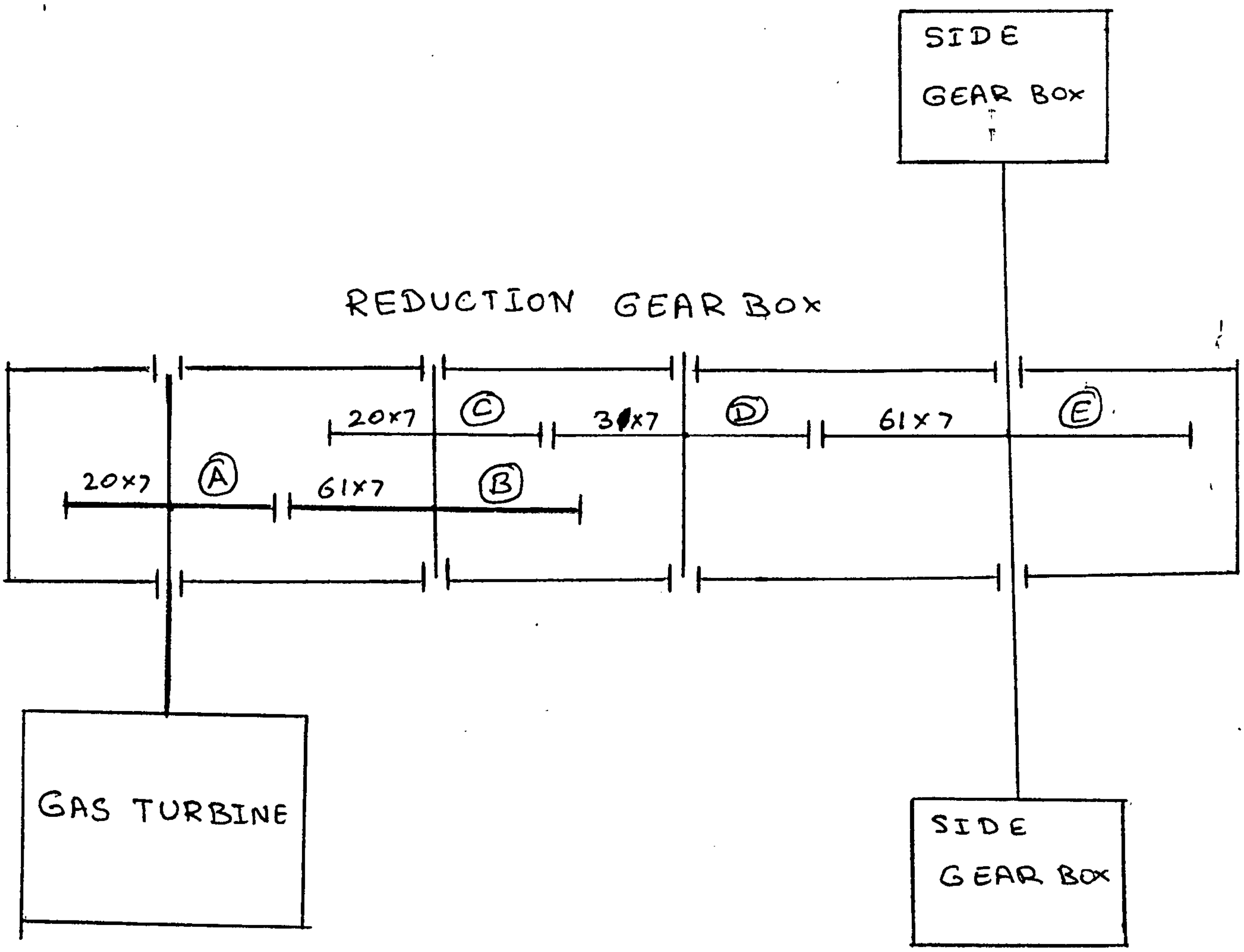


Fig-29.

(d) I stage.

Module	=	7	
No. Of teeth on pinion	=	20	(A)
No. Of teeth on gear	=	61	(B)
Reduction ratio at I stage	=	61/20	
	=	3.05	

(e) II stage.

Module	=	7	
No. Of teeth on pinion	=	20	(C)
No. of teeth on gear	=	61	(E)
No. Of teeth on idler gear	=	31	(D)
Reduction ratio	=	61/20	
	=	3.05	
Over all reduction ratio	=	3.05 X 3.05	
	=	9.3025	

The working distance is 784mm any adjustment in spacing if required various permutations and combinations are possible and can be increased or decreased by adding inter mediate gears also but the direction of rotation is to be kept in mind.

201. The material selected for the gear is Nickel Chromium steel and the nomenclature is 30 Ni 4 Cr 1

The composition is

Carbon	-	0.16 to 0.22
Silicon	-	0.17 to 0.37
Mn	-	0.3 to 0.6
Cr	-	1.25 to 1.65
Ni	-	3.25 to 3.65
P	-	0.025
S	-	0.025
Yield point	-	110 kgf/sq. mm
Ultimate TS	-	130 kgf/sq. mm
Elongation	-	9
Impact strength	-	8

Bending stress	-	35 kgf/sq. mm
Contact stress	-	180 kgf/sq. mm
Maximum torque	-	440 kgf-m at 1800 RPM

202. Program to evaluate tooth proportions.

I stage

	Pinion	Gear
Module	7	7
No. Of teeth	20	61
Pressure angle	20	20
Profile correction	0.000	0.000
Pitch cir rad (theo)	70.000	213.000
Pitch cir rad (work)	70.000	213.000
Base cir rad	65.779	200.625
Working pressure angle	20.000	20.000
Centre distance (theo)	283.500	283.500
Centre dist (work)	283.500	283.500
Addendum cir rad	77.000	220.500
Root cir rad	61.250	204.750

Intermediate stage

	Pinion	Gear
Module	7	7
No. Of teeth	20	31
Pressure angle	20	20
Profile correction	0.000	0.000
Pitch cir rad (theo)	70.000	108.500
Pitch cir rad (work)	70.000	108.500
Base cir rad	65.779	101.957
Working pressure angle	20.000	20.000
Centre distance (theo)	178.500	178.500
Centre dist (work)	178.500	178.500
Addendum cir rad	77.000	115.500
Root cir rad	61.250	99.750

II stage

	Pinion	Gear
Module	7	7
No. Of teeth	31	61

Pressure angle	20	20
Profile correction	0.000	0.000
Pitch cir rad (theo)	108.500	213.500
Pitch cir rad (work)	108.500	213.500
Base cir rad	101.957	200.625
Working pressure angle	20.000	20.000
Centre distance (theo)	322.00	322.000
Centre dist (work)	322.000	322.000
Addendum cir rad	115.500	220.500
Root cir rad	99.750	204.750

203. To calculate varioud stresses i.e. comming on the gear, torque transmitted, cutter dimensions etc few factors are required to be calculated and fed to the software that is available in Fortran IV for calculation of gear dimensions are.

<u>I Stage.</u>

RPM of turbine	-	16,754
Torque	-	47 kgm
No. Of teeth	-	20

Module		-	7
Surface velocity		-	$Vf = \frac{Pi \times 20 \times 7 \times 16754}{25.4 \times 12}$
		=	241758
Velocity factor	Cv	=	$\sqrt{78}/(78 + \sqrt{Vf})$
		=	0.578
Vf (61)		=	7926.502
Cv (61)		=	0.6833
Torque		=	143.289

II stage

Vf	=	5113.87
Cv	=	0.7228
Torque (61)	=	437 kgm

204. Dimensions.

DATA	Pinion Dimensions	Gear Dimensions
No. of teeth	20	61
Module	7.000	7.000
STD pressure angle degrees	20.00	20.00
Profile modification XM	3.656	0.000

Standard centre distance MM	283.500	283.500
Working centre distance MM	287.000	287.000
Addendum circle radius MM	80.500	220.344
Root circle radius MM	64.906	204.750
Cutter edge radius MM	1.400	1.400
Face width MM	25.00	25.00
Maximum torque kgm	47.00	143.00
Over load factor ko	1.000	1.000
Size factor ks	1.000	1.000
Dynamic factor kv	0.578	0.578
Load distribution factor km	1.300	1.300
Elastic coefficient cp	60.748	60.748
Size factor cs	1.000	1.000
Surface condition factor cf	1.000	1.000
Over load factor co	1.000	1.000
Dynamic factor cv	0.578	0.578
Load distribution factor cm	1.300	1.300

CHECK VALUES

Tooth fillet radius MM	2.100	2.100
OL in MM	79.892	223.326
YF in MM	72.692	214.626
Film in degrees	62.326	66.581
X in MM	7.623	7.592
T in MM	15.246	15.185
H in MM	7.335	8.829
Ratio of LM/(2 * LR)	1.009	1.007

RESULT.

Working pressure angle degrees	21.839	21.839
Base circle radius MM	65.778	200.624
STD pitch circle radius MM	70.000	213.500
Working pitch circle radius MM	70.864	216.136
Load angle degrees	25.190	20.810
Stress correction factor	1.932	1.793
Load sharing ration MN	1.00	1.00
Tooth form factor Y	0.925	0.693
Geometry factor (bending) J	0.478	0.386
Geometry factor (contact) I	0.09094	0.09094
Contact ratio	1.488	1.488
Bending stress kgf/sq. mm	17.81	22.05
Contact stress kgf/sq. mm	130.68	130.68

DATA	Pinion Dimensions	Gear Dimensions
No. of teeth	20	31
Module	7.000	7.000
STD pressure angle degrees	20.00	20.00
Profile modification XM	3.959	3.959
Standard centre distance MM	178.500	178.500
Working centre distance MM	185.500	185.500
Addendum circle radius MM	80.041	118.541
Root circle radius MM	65.209	103.709
Cutter edge radius MM	1.400	1.400
Face width MM	40.00	40.00

Maximum torque kgm	143.300	222.100
Over load factor ko	1.000	1.000
Size factor ks	1.000	1.000
Dynamic factor kv	0.683	0.683
Load distribution factor km	1.300	1.300
Elastic coefficient cp	60.748	60.748
Size factor cs	1.000	1.000
Surface condition factor cf	1.000	1.000
Over load factor co	1.000	1.000
Dynamic factor cv	0.683	0.683
Load distribution factor cm	1.300	1.300

CHECK VALUES

Tooth fillet radius MM	2.100	2.100
OL in MM	82.857	122.421
YF in MM	74.357	113.421
Film in degrees	65.879	66.289
X in MM	7.652	7.866
T in MM	15.304	15.732
H in MM	8.590	8.910
Ratio of LM/(2 * LR)	1.005	0.995

RESULT.

Working pressure angle degrees	25.280	25.280
Base circle radius MM	65.778	101.957
STD pitch circle radius MM	70.000	108.500
Working pitch circle radius MM	72.745	112.755
Load angle degrees	27.794	25.983

Stress correction factor	1.632	1.630
Load sharing ratio MN	1.00	1.00
Tooth form factor Y	0.787	0.777
Geometry factor (bending) J	0.482	0.476
Geometry factor (contact) I	0.10071	0.10071
Contact ratio	1.300	1.300
Bending stress kgf/sq. mm	27.76	28.09
Contact stress kgf/sq. mm	153.60	153.60

DATA	Pinion Dimensions	Gear Dimensions
No. of teeth	31	61
Module	7.000	7.000
STD pressure angle degrees	20.00	20.00
Profile modification XM	3.959	0.000
Standard centre distance MM	322.000	322.000
Working centre distance MM	325.500	325.500
Addendum circle radius MM	119.000	220.041
Root circle radius MM	103.709	204.750
Cutter edge radius MM	1.400	1.400
Face width MM	40.000	40.000
Maximum torque kgm	222.100	437.000
Over load factor ko	1.000	1.000
Size factor ks	1.000	1.000
Dynamic factor kv	0.722	0.722
Load distribution factor km	1.300	1.300
Elastic coefficient cp	60.748	60.748
Size factor cs	1.000	1.000
Surface condition factor cf	1.000	1.000
Over load factor co	1.000	1.000

Dynamic factor cv	0.722	0.722
Load distribution factor cm	1.300	1.300

CHECK VALUES

Tooth fillet radius MM	2.100	2.100
OL in MM	118.373	222.026
YF in MM	111.373	213.926
Film in degrees	60.467	64.932
X in MM	7.961	7.617
T in MM	15.923	15.234
H in MM	7.053	8.185
Ratio of LM/(2 * LR)	1.004	1.005

RESULT.

Working pressure angle degrees	21.629	21.629
Base circle radius MM	101.957	200.624
STD pitch circle radius MM	108.500	213.500
Working pitch circle radius MM	109.679	215.821
Load angle degrees	23.730	20.311
Stress correction factor	2.021	1.862
Load sharing ratio MN	1.00	1.00
Tooth form factor Y	1.041	0.756
Geometry factor (bending) J	0.515	0.406
Geometry factor (contact) I	0.09658	0.09658
Contact ratio	1.537	1.537
Bending stress kgf/sq. mm	25.26	32.07
Contact stress kgf/sq. mm	125.98	125.98

204. The final dimensions of gear with various correction for different stages are tabulated below.

I stage

	Pinion	Gear
Module	7	7
No. Of teeth	20	61
Pressure angle	20	20
Profile correction	3.656481	0.000
Pitch cir rad (theo)	70.000	213.000
Base cir rad	65.779	200.625
Centre distance (theo)	283.500	283.500
Centre dist (work)	287.000	287.000
Addendum cir rad	80.500	220.344
Root cir rad	64.906	204.750

Intermediate stage

	Pinion	Gear
Module	7	7
No. Of teeth	20	31
Pressure angle	20	20

Profile correction	3.95939	3.95939
Pitch cir rad (theo)	70.000	108.500
Base cir rad	65.779	101.957
Centre distance (theo)	178.500	178.500
Centre dist (work)	185.500	185.500
Addendum cir rad	80.041	118.540
Root cir rad	65.209	103.709

<u>II stage</u>

	Pinion	Gear
Module	7	7
No. Of teeth	31	61
Pressure angle	20	20
Profile correction	3.95939	0.000
Pitch cir rad (theo)	108.500	213.500
Base cir rad	101.957	200.625
Centre distance (theo)	322.00	322.000

Centre dist (work)	325.500	325.500
Addendum cir rad	119.000	220.040
Root cir rad	103.709	204.750

<u>Analysis of design.</u>

205 The gears are designed for infinitelife where the life factor LH for a cycle of life more than 10^8 the value of LH is 1.00. In practice infinite life can not be achieved however, this is likely to give maximum life.

206. The permissible bending stress for 30 Ni 4 Cr1 steel is 35 kgf/sq. mm where as the calculated bending stress is 32 kgf/sq. mm and the permissible contact stress is 180 kgf/sq. mm which is also within the permissible limit.

207. The achieved centre to centre distance of the arrengement of gears is 797 if arranged in straight line, the required centre to centre distance of 784 mm can be achieved by staggered arrangement of gears. The length of the reduction gear box will be approximately 1050 mm and can suitably accommodated by removing the present transmission gear unit.

ADDITIONAL ADVANTAGES

208. There are some additional advantages which were not discussed earlier deliberately are discussed here now.

(a) **<u>No Cooling water.</u>** All gasturbines are air cooled. For this purpose some amount of compressed air is taken from the ouput stage of compressor directly to the turbine blades no cooling water is used and hence freedom from water troubles like leakage, cooling, boiling, wastages erosion (when salt water is used) .

(b) **Low vibration.** As all mechanical motion is rotary, vibration produced by gas turbine is very small. The static vibration of gas turbines is 1/4 that of diesels, and the dynamic weight of gas turbine is less than 10% of the static weight while it is about 50% for diesels. Because of this the foundation requirement is not so heavy.

(c) **Low noise.** The gas turbine engine rotates at high speed, noise produced is at high frequency and is easy to attenuate. So the acoustic enclosure is simple in construction while that for diesels is much heavier. .

(d) **Compact and light weight.** Gas turbines are one fourth in weight andone seventh in volume as compared with diesels.

(e) **Easy operation.** With development of electronics the control system is made of electonics. This facilitates automatic adjustment of fuel air ratio by changing the initial setting at control panel itself. So practically the engine can operate in any kind of fuel that is available on the earth.

(f) **Clean exhaust gas.**: Due to perfect combustion of fuel oil in gas turbines, Exhaust gas is much clean. A typical example shows that NOX is 1/10, CO is 1/40, so x is 1/5 in volume less than that of diesel emissions on NO2 diesel oil. Further more, Carbon powder (as black smoke) is hard to see and due to its construction that lube oil is completely seperated from the hot part, lube oil is not mixed into the exhaust gas passage. When an optional gas fuel system is employed, the exhaust gas will be much cleaner. The future is becoming more important in keeping our enviornment clean. It is possible to reduce NOX further by using an optional water injection system.

(g) **Low maintenance cost.** Diesel engines require weekly maintenance while gas turbines need only a few minutes operation at intervals of two months. Maintenance run can be done at no-load, while Diesel engines need on load run. Simple consturction of gasturbine is easy to maintain as no of construction parts are only 20% - 40% of diesels. Maintenance parts are also light in weight. When required to be removed it is quite easy as it consumes less time. For e.g. AGT 1500 i.e fitted in XMI can be removed within 25 minutes where as removal of present engins V-46-6 i.e. fitted in T-72 takes about 12 hours.

(h) **Low lube oil consumption.** Lubrication oil consumption of gas turbine is only 0.08 l/hr at 1000 KW class as compared to that of 8 gms/hp/hr in case of present V-46-6 diesel engine more over the frequency of oil change can be reduced to considerable extent as lubrication oil in case of gas turbine is not subjected to such high temperature and pressure as compared to that of Diesel engine.

(j) **Easy inspection.** The gas turbine can be easily inspected without disassembly with the help of a bore scope where as it is not possible in case of diesel engines.

(k) **Fast loading time.** The gas turbine can be loaded with 100% rated load immediately after starting. Where as in case of diesel engine warming up is required before loading.

(l) **Absorbs large reverse power.** As gas turbines have large moment of inertia large reverse power produced in case of an AFV is absorbed without any damage to the engine. This capacity is estimated as much as 4 times that of diesels.

(m) **Operates on a variety of fuels.** Can operate on any kind of fuel with a suitable electonic control system the fuel variation can be carried out at control panel itself where as adjustment to FIP is required in case of diesel engines. There is no complex mechanism which are required to advance or retard the fuel injection like as in case of diesel engine. A dual fuel system is also possible (like liquid and gas fuel).

(n) **High initial starting torque .** The starting torque of a gas turbine is high and can be compared to that of a DC motor of same power rating. The torque increases if the going gets difficult and there is no optimal torque as it occurs in case of Diesel.

COMMENTS ON FEASIBILITY OF FITMENT

209. It is feasible to fit the selected gas turbine along with the designed reduction gear box along with the air cleaner in the available space of engine compartment. The layout of the assembly is as shown in the Figs-30 & 31 in both configuration i.e. with complete mechanical arrengement and possible electrical arrengement.

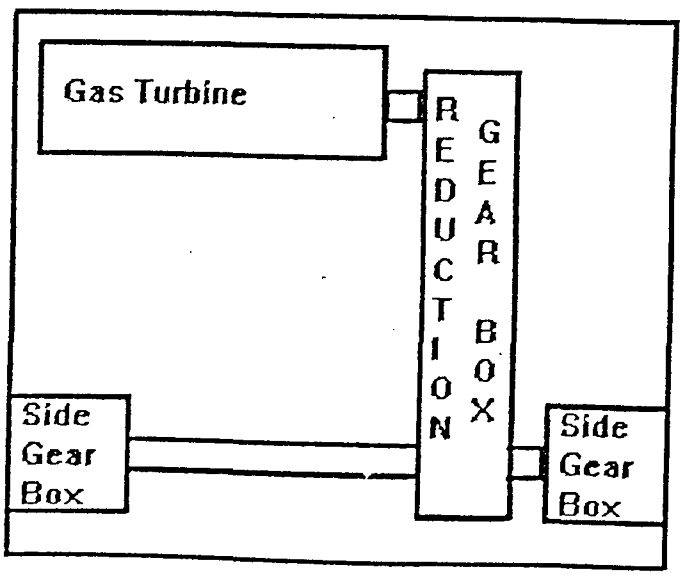


Fig-30.

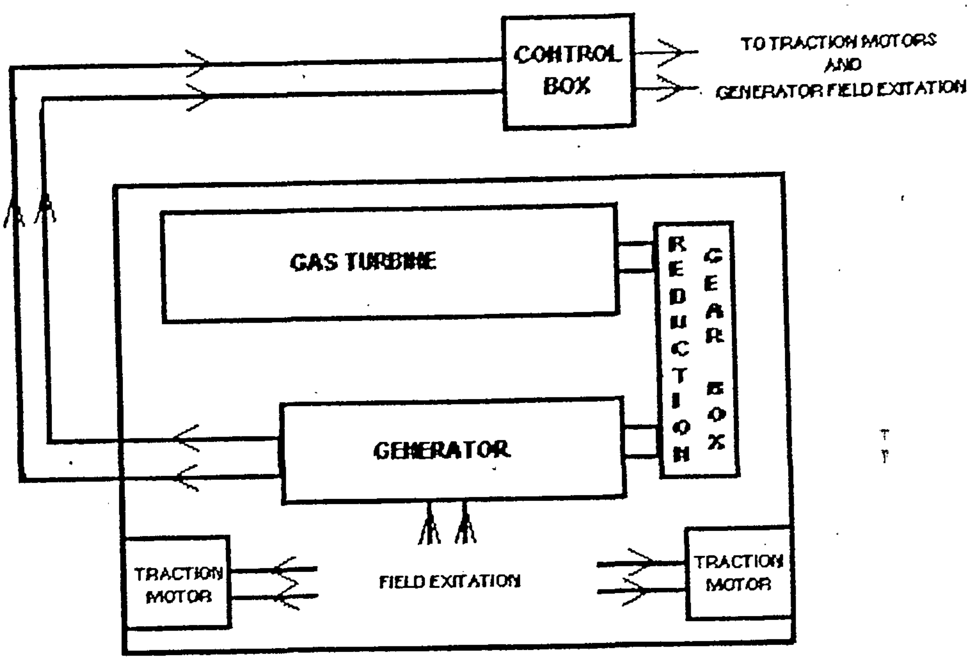


Fig-31.

LATEST DEVELOPMENTS

210. It is believed that the diesel engine has reached its ultimate of development and gas turbines are still in the initial stages as far as vehicular applications are considered. USA has awarded contract for development of gas turbine for tank application under AIPS-T (Advanced integrated propulsion systems Turbines) to General Electric a trubine major for a futuristic power pack by US Army Tank Automotive Command (TACOM) which has resulted in LV100 power pack. The target that was set was to demonstrate the potential for advanced technology to provide desired combat capability in greatly reduced volume, improved fuel economy and with reduced sustainment costs for tracked combat vehicles.

211. The gas turbine power unit is expected to provide specific advantage over diesel engines no smoke, low noise, true multifuel capability with no adjustments and best available low speed torque charecteristic. Low fuel consumption charecteristic for battle field and peace time operation is provided by a variable area compressor and turbine to gain efficient operation over a wide power range, a digital control system to fully integrate power pack systems functions.

212. The AIPS contract set goals and present LV 100 status are tabulated below:

AIPS Contract Goals	LV 100 Status
Power pack volume 194 cu.mtr (with one battle field day fuel)	Less than 175 cubic mtr
1070 HP at sprocket	Exceeds requirement
Accelaration 0-20 mph in 7 secs	Beats requirements
Continous downhill braking	With cooling margin beyond required 30%
Reduced fuel consumption	80% Range improvement over XM1
0.12 maintenance manhours/op hour	Better than requirement
1000 mean km between failures	Exceeds by over four times
1500 mean km between failures (mission)	Exceeds by over five times
Fuel tolerance	DF1/2, JP-8, gasolene - no adjustments required.
Low weight	1 1/2 tons lighter than diesel

213. Though the claims made are very high nothing specific regarding fuel consumption are known. However, it can be understood that with a wide experience gained on AGT 1500 systems the improvements made on LV 100 power pack may be very competitive as compared to that of AGT 1500 or any other diesel engine of similar power rating. But nothing much is publicised. The publicity claims are very ambitious for example the specific fuel consumption of MTU engine i.e. fitted on MBT Arjun was around 174g/HP/hr where as the test results show that it consumes 194 g/hp/hr a difference of 20g/hp/hr and hence nothing much can be made out on the claims made by GE on the performance of LV 100.

CONCLUSION

214. So far most of the aspects of gas turbines have been discussed, but yet the gas turbine has not emerged as a competitor to diesel engine mainly because of the cost and fuel consumption, also the argument of increased size of air cleaner which is three times as much as that of diesel engine in the same power range and coupled with this the increased quantity of fuel i.e. required for same range the gas turbine unit occupies more specific volume than that of diesel engine in the same power range increases the overall weight of the AFV because of additional armour protection i.e. required to be provided. However, the cost factor I have my own comments the present MTU power pack costs about Rs.Five Crores, the AGT power pack costs about Rs.Four Crores and the uprated T-72 engine along with transmission gear unit and side gear boxes costs about Rs.Sixty Lakhs which is a wide margin. Latest innovative development of rotating type drum air cleaner has reduced the size of the air cleaner to sixty percent of the earlier size coupled with transverse mounted engine has saved a space of 41% which has reduced the specific volume of the AGT 1500 power pack.

215. The development of LV 100 gas turbine power pack with variable area turbine nozzle (VATN) the specific fuel consumption and specific vloume of the whole unit has reduced to a great extent, the gas turbines have come a long way in tank applications. Most likely the LV 100 is likely to emerge as potential competitor to that of AIPS diesel version. The future seems to be bright for gas turbines as adequate research effort is going on in this field. The tank engine is the most important factor in deciding all other factors of the tank i.e. the degree of armour protection and the fire power that can be achieved. This can be adequately summed up that the engine is not only a power producing unit but also a potent weapon. With this foresight Gen Guderian the II world war fame Nazi General has quoted

> "Engine of a tank is no less a weapon
> as compared to that of the gun".

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