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SIX MILLION MILES EXPERIENCE WITH GAS TURBINE LOCOMOTIVES

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In February, 1956, just four years after the Union Pacific's first gas turbine-electric locomotive was put to work, the turbine fleet rounded out 6 million miles of service. During this period, the 25 "Big Blows" have delivered a performance record which amply justifies the Union Pacific decision to pioneer this new type of motive power.

Overall service totals accumulated during the years 1952-1955 are shown in Fig. 1. This chart does not include figures for the seven months operation of one locomotive fueled with propane gas as an experiment.

During this period, the 27 turbine power plants, including 2 spare turbines, have accumulated 227,950 fired hours. Maximum hours for one turbine are 17,266 hours as of June 1, 1956, on the plant originally delivered with locomotive No. 55 in July 1952.

In Fig. 2 are tabulated performance averages based on service totaled for the four-year period.

Most of these figures are difficult to compare if one is not closely acquainted with railroad operation. As an indication, the miles per month are about average for modern mainline freight service, the train miles per hour are very good and it is expected that the fuel consumption figures would be startling to one accustomed to piston engine data. Imagine the reaction to a new sport car getting 360 ft per gal at 33 mph average. As a gage of their effectiveness, the 25 locomotives handled approximately 10 per cent of the total freight traffic on the Union Pacific during 1955. During this period they moved 38.5 per cent of the tonnage on their assigned Division.

The auxiliary diesel fuel shown in the table is consumed by the turbine when starting or shutting down, by the auxiliary diesel engine when the turbine is not running and by the steam generator used to heat heavy fuel.

Development Program

Both General Electric Company and the Union Pacific realized at the start of this program that the locomotives were far from perfect. Therefore, a development program was initiated with the first locomotive and has been continued throughout the period. Improvements were made at the factory when possible and on the railroad on locomotives in service. Up to the present time, there have been approximately 319 authorized modifications which applied to one or more locomotives. These have been progressed in a scheduled program to suit railroad traffic requirements.

As with most carefully scheduled programs, there have been some unforeseen developments requiring emergency action to keep freight trains rolling on time across turbine territory, the Wyoming Division.

Fuel Development

One of the more outstanding and probably the most important reason for gas turbine success on the railroad has been the ability to keep the locomotives supplied with a relatively low-cost residual fuel blended or treated to suit the turbine limitations.

The single shaft, open cycle, non-regenerative gas turbine as we now know it is limited to a full load thermal efficiency of approximately 17%. This can be improved by increasing turbine inlet temperature above 1300°F. At present, neither alloys nor ceramics are readily available for service above this temperature, or even up to this temperature, with standard Bunker C fuels.

A further obstacle to be overcome is the load cycle, Fig. 3. In railroad service, this depends to some extent on the track profile and varies with train tonnages and traffic conditions. Records show that the turbine load factor based on total fired hours is just under 50%. This differs from the diagram because it includes idle time at terminals and a large amount of development testing.

Machine efficiencies for turbines and compressors have been pushed up to approximately 86%. Small gains can be expected, but they will be small and hard to come by. Regenerative cycles are promising, but an effective heat exchanger on paper may prove to be impossible to fit into a locomotive or impossible to maintain on a reasonable basis.

The machine limitations leave one avenue open: simply develop plentiful sources of cheap specification fuel.

Briefly, the builder's specification for gas turbine fuel oils is as follows:

1. <u>Viscosity:</u> 95 SSU at nozzles with max. temp. of 250°F satisfying stability limit of Item 3.

2. <u>Character of Ash:</u>
   (a) Weight ratio of sodium in ash to Vanadium in ash not greater than 0.3.

   (b) Weight ratio of magnesium in ash to Vanadium in ash not less than 3.0.

   If Vanadium content is less than 3 ppm, magnesium ratio need not be followed.

   (c) Sodium content of oil should not exceed 10 ppm.

   (d) Calcium content should not exceed 10 ppm.

   (e) Total ash content should not exceed 2,000 ppm.

   (f) Additives must be soluble or finely ground to prevent settling before use.

3. <u>Stability:</u>
   Below 210°F. - No. 2 tube or better, Sepc. MIL F-859.
   Above 210°F. - No. 1 tube or better.

4. <u>Compatibility:</u> No. 2 or better in the NBTL heater test with other fuels used.

5. Bottom solids and water shall not exceed 2%.

6. Sulphur content shall not exceed 3-1/2%.

This specification is quite restrictive when compared to ordinary No. 6 or Bunker C fuel. Eventually our gas turbine must digest a No. 6 grade fuel oil, because it is very advantageous to have unrestricted access to fuel stocks of various refiners along the railroad and to obtain such fuel at lowest market price. Since the basic thermal efficiencies are fixed for any particular load factor, it is necessary that fuel costs be controlled in order to make turbine engines fulfill their promise. As shown in Fig. 4, the economical limit for cost of turbine fuel is reached when turbine fuel price becomes 34% of the price of diesel fuel. This is based on a turbine load factor of 50%.

This points up another factor which must be considered with gas turbine locomotives in their present form. They are strictly a main line power plant which must be kept working at as near 100% full load as possible.

So far, the Union Pacific has been able to supply gas turbines with fuel at costs which keep them in competitive position. In the future, this will be more difficult but there is still the cushion of costs for maintaining a basically simple non-reciprocating, single unit locomotive compared to the cost of multiple unit reciprocating engine locomotives.

With the 25 gas turbines burning approximately 75,000 barrels of fuel per month, the testing and selection of a suitable fuel has been no small project. In this respect, the Union Pacific has been fortunate in having access to very co-operative on-line and West Coast refineries.

Table No. I shows an analysis of some of the fuels used for turbine fuel.

Fuel No. 1 was a Bunker C fuel untreated. This fuel was used in 1949 and 1950 on the General Electric experimental locomotive; it caused ash deposit and corrosion on turbine nozzles and buckets.

Fuel No. 2 was a low ash fuel made by pulling No. 1 fuel overhead, leaving the ash in the still bottoms.

Fuel No. 3 was fuel No. 2 with oil soluble calcium added to neutralize the corrosive action of vanadium found in General Electric laboratory work. It was abandoned because of the 15¢ to 25¢ per barrel premium over Bunker C.

Fuel No. 4 was heavy overhead distillate which gave excellent results but was discontinued because of a 55¢ per barrel premium cost.

Fuel No. 5 represented a new approach to using Bunker C. This fuel was made from selected low sodium crude, de-salted by washing and with calcium napthenate added to provide a ratio of 5 ppm calcium to 1 ppm vanadium. This fuel was used with good results for approximately a year. In October, 1953, continued work on corrosion by G.E. showed that the calcium additive was responsible for some corrosion at temperatures below 1300°F. Magnesium which showed better characteristics was then substituted for the calcium. Magnesium was added to provide a ratio of 3 parts magnesium by weight to 1 part vanadium by weight.

This became turbine fuel No. 6, known on the railroad as "Bunker M" or M fuel. The changeover to M fuel resulted in operating difficulty about 2 weeks after initial stocks were mingled in storage. Locomotive fuel filters began to plug rapidly and it was discovered that the mixed fuel viscosity had jumped from 200 SF @ 122°F to 260. As an emergency measure, filter elements were removed until the stored fuel viscosity could be reduced by adding approximately 20% fuel No. 8, a Wyoming cat cycle cutter stock. No. 8 fuel, being a No. 2 burner fuel was a good fuel by itself, but the cost was too great and sufficient stocks were not available.

Fuel No. 7 was a Montana cat cycle stock tested in one locomotive with excellent results but a 65¢ per barrel premium ruled it out.

Fuel No. 10, started in the summer of 1954, taught another lesson. The heavy overhead fuel distillate from deeply fluid cracked stocks had a furol viscosity of 190 secs @ 122°F and an API gravity of 5.6. Immediately an epidemic of fuel pump failures caused by stuck pistons began. Frantic measures such as solvent additive, increased piston clearance and modified pump flushing circuits were of little value. In order to get out of trouble, it was necessary to add 10% cutter stock at the refinery, reducing the viscosity to 63 SF @ 122°F. This became fuel No. 11. Being very low in ash content made it an ideal fuel from the standpoint of corrosion.

Since large quantities of approximately 200 furol fuel had previously been handled without difficulty, it became apparent that there is a relationship between gravity and viscosity which cannot be ignored in selecting fuels for this type of system.

Until the latter part of 1955, the viscosity of turbine fuels was held at 35-40 furol with a cost penalty of 15¢ to 21¢ per barrel.

Fuel No. 9 was similar to fuel No. 11, with a cost penalty of 25¢ to 30¢ per barrel.

In order to reduce the cost and increase the supply, Fuel No. 12 was made by blending Fuel No. 9 fifty-fifty with a calcium-free residual. This required use of magnesium additive but resulted in reducing the cost penalty to 21¢ per barrel.

In order to overcome the viscosity limitations of the fuel pump, a new fuel metering system was developed. Fuel No. 13 is a test fuel designed to test the performance of the pump. It was made by blending tar bottoms fifty-fifty with fuel No. 9 plus magnesium additive. This worked well for two months, when a change in base fuel at the refinery resulted in fuel atomizing nozzles plugging. The fuel metering system withstood the test.

Fuel No. 14 is a Wyoming Bunker C type fuel with magnesium additive. Availability varies during the year since sodium content of the crude fluctuates and since de-salting facilities are not available, there are times when the fuel cannot be used for turbines.

Fuel No. 15 is typical of that now being shipped by our principal supplier. It requires no additive but can be cut back to lower viscosities if fuel pump trouble should recur.

Turbine Development

Paralleling the fuel development there has been an intensive program for mechanical improvement. Refer to Fig. 5. Most of the modifications have been tried out in G.E. laboratories before field testing. Some of the major problems accomplished:

1. Redesign of second stage turbine buckets to eliminate fatigue failure. This involved change in bucket shape and using a material having better internal damping characteristics. Turbine speed control was also modified from seven steps tied to throttle to 2 steps, idle and top speed controlled by transition selection. This reduced the number of times the rotor passed through minor criticals during normal operation.

2. Fuel nozzle improvements resulting in elimination of pintle type nozzle have pushed nozzle life between overhauls from 50 hours to 400 hours. This has also resulted in increased combustion chamber liner life by reducing damage from uneven flame patterns.

3. Combustion chamber liner life through constant development and experimentation has been increased from less than 200 hours to an average of well over 2,000 hours. At the same time, inspection and assembly techniques have improved to the point where only chambers found faulty during a borescope inspection need be disassembled. This reduces chamber renewal time from 8 hours to one or two hours. Some of the earliest inspections required 48 to 60 hours due to difficulty in repairing cracked and warped parts.

4. Locomotives 51 to 57 (refer to Fig. 6) were originally constructed with air intakes inside the engine room with turbine inlet air passing through viscous impingement type carbody filters. It was found that there was considerable power loss due to hot air entering the intake from the engine room and relatively rapid loss of compressor efficiency due to oil and dirt building up on compressor blading. This was eliminated by moving air inlets to the roof with no filtering. Refer to Fig. 7. The increased compressor wear due to particle abrasion appears to be minor.

5. The turbine lubrication system has a serious fault. Normal lubricating oil circulation is provided by an AC motor driven pump. A DC driven pump controlled by pressure switch provides lubrication and cooling when the AC pump is not functioning. There have been several cases when the babbitt bearings have failed, due to electrical failure of one or both pumps. A mechanical drive pump for the main oil source would have prevented most of the failures but to date a satisfactory mechanical drive has not been developed due to equipment arrangement problems.

6. Originally, turbine inlet temperature limit was controlled by a gas filled bulb in the exhaust stack. Loss of gas pressure removed the temperature limit from the turbine with no warning except a suddenly powerful locomotive. Refer to Fig. 8. This was not always recognized in time, resulting in damage to turbine buckets.

A new temperature control device which operates on air pressure regulated by differential expansion thermostats in the exhaust stack is designed to fail safe and will shut the turbine down in event of control failure.

Air pressure shutdown devices controlled by fusible plugs were used in the interim period, but proved unreliable.

7. Turbine bucket life has been fair with the corrosion and overtemperature problems (see Fig. 8) now pretty well under control. A recent modification to reduce fatigue stress on first stage buckets caused by thermal shock is expected to increase resistance to fatigue cracking approximately 9 to 1. This change entails recontouring the leading edge to eliminate the thin edge, thus getting more even expansion, and reducing fuel rate during turbine starts to give bucket temperatures a longer time to stabilize.

8. The present fuel metering pump is an 18-cylinder, variable stroke, wobble plate pump with piston return by hydraulic pressure. Three pistons 120° apart deliver fuel to each of the six combustion chambers. Pump life is currently very good. Piston sticking with the very tarry fuels has not been overcome and fuel temperatures must be kept below approximately 230°F to avoid pistons sticking from distortion. This again limits the viscosity of fuels below the specification temperature of 250° to get 95 SSU at the nozzles.

A fuel metering system using hydraulically driven gear pumps for flow division has been worked out and operated on a test basis.

<u>Limitations</u>

The gas turbine power plant has some very definite limitations when applied to railroad locomotive service. As previously mentioned, even main line freight service contains a lot of low load and idle time. This means that the machine must be specially designed to idle at low speeds and low fuel rates. A simple cycle machine of the type now in use requires approximately two-thirds of the gross output of the turbine at full load to drive the compressor. This amounts to roughly 10,000 hp on a machine rated at 5000 shaft hp. The compressor load does not fall off rapidly with reduced speed in the range used and at top speed no load remains at the full 10,000 hp.

The second limitation is the extreme fluctuation between net power available at low and high altitudes and between low and high ambient temperatures. This has led the Union Pacific to specify that power available for traction be based on 90°F ambient and 6,000 ft altitude for future machines.

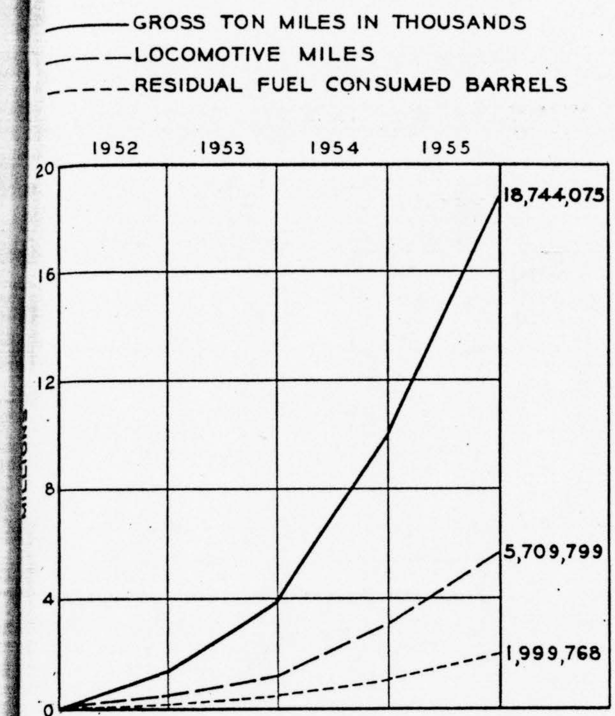


FIG. 1 GAS TURBINE LOCOMOTIVE PERFORMANCE TOTALS

LOCOMOTIVE MILES PER MONTH	10,1[illegible]
TURBINE HOURS PER MONTH	403.45
TRAIN MILES PER TRAIN HOUR	33.10
GROSS TON MILES PER TRAIN HOUR	112,237
GROSS TONS PER TRAIN	3391
RESIDUAL FUEL PER LOCO MILE GAL.	14.71
RESIDUAL FUEL PER TURBINE HOUR GAL.	368.46
RESIDUAL FUEL PER 1000 GTM GAL.	4.690
AUX. DIESEL FUEL PER LOCO MILE GAL.	1.040
AUX. DIESEL FUEL PER TURBINE HOUR GAL.	26.04
AUX. DIESEL FUEL PER 1000 GTM GAL.	.317

FIG. 2 GAS TURBINE LOCOMOTIVE PERFORMANCE AVERAGES 1952 - 1955

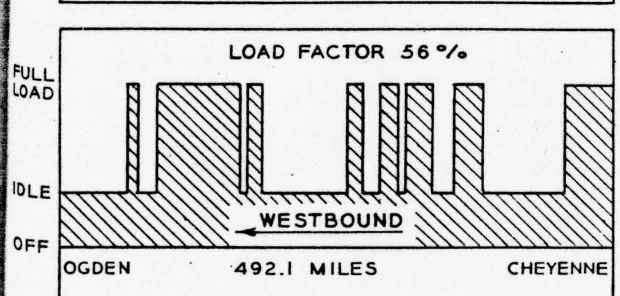


FIG. 3 TURBINE LOAD CYCLE ESTIMATED FROM TRACK PROFILE

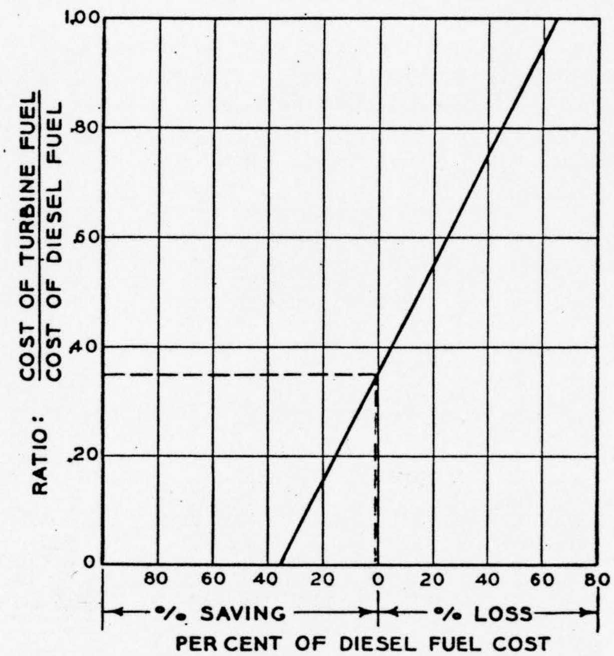
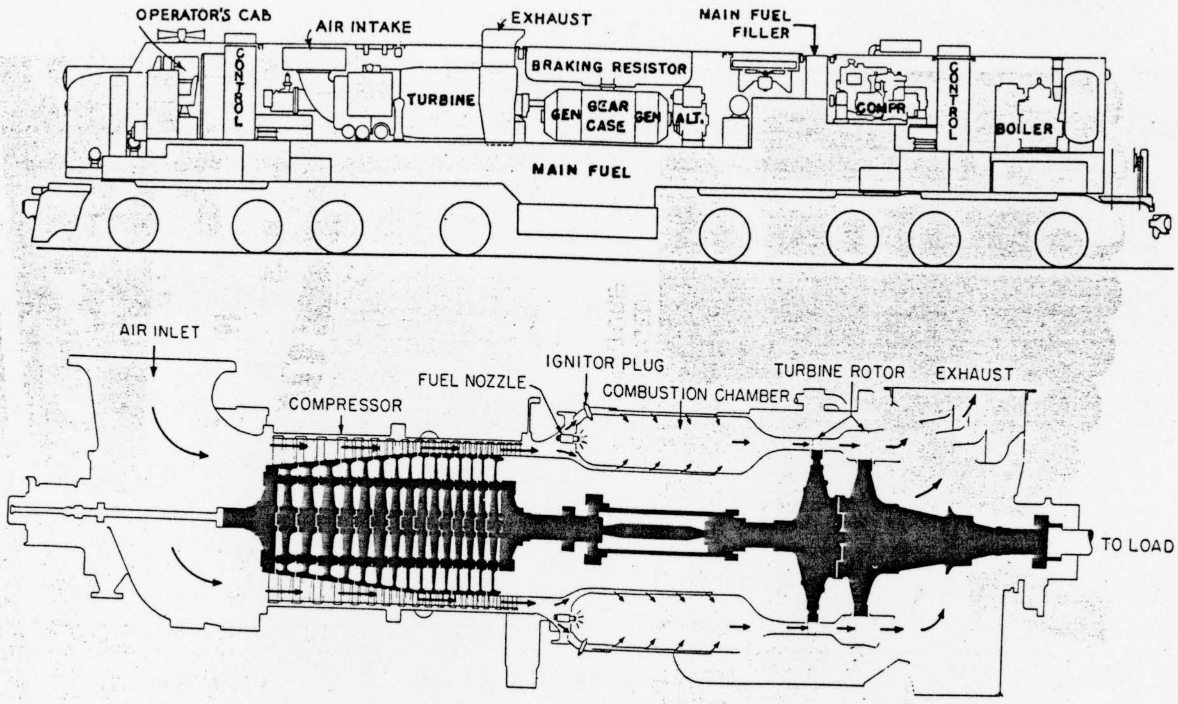


FIG. 4 RELATIONSHIP BETWEEN COST OF TURBINE LOCOMOTIVE FUEL AND DIESEL LOCOMOTIVE FUEL AT 50% LOAD FACTOR

HISTORY OF UNION PACIFIC GAS TURBINE FUEL															
	1 #6 CALIF.	2 LOW ASH CALIF.	3 #2 LOW ASH CALIF. +CA	4 O.H. DIST. CALIF.	5 #6 D.S.+CA CALIF.	6 #6 D.S.+MG CALIF.	7 CAT.CYCLE DIST. MONT.	8 CYCLE STOCK WYO.	9 O.H. HVY.DIST. WYO.	10 FLUID CRACKED CALIF.	11 #10 PLUS CUTTER	12 #6 SPECIAL WYO.	13 SPEC. TEST WYO.	14 #6 WYO. +MG	15 FLUID CRACKED CALIF.
GRAV. API	8.2	8.1		11.4	7.8	7.7	17.7	7.7	6.3	5.6		10	10.6	7	-1.0
SULFUR %	1.25	1.25		1.24	1.18	1.34	2.57	3.17	3.10	1.14		2.5	3.0	2.7	1.0
FLASH PM	190	280		220	185	176	250	285	130	240	194	150	195	235	325
FLASH COC						210			265	355	250		220		
VIS. SUS @ 100° F.				182			40	62							
VIS. SF @ 122° F.	200	200		10	190	194			14.8	190	63	35-40	195-200	150	85-100
POUR POINT °F.	+35	+85		+65	+25	+30	+30	-15	+53	+40	+35	+30		+45	
CAR.RES. %	17.5	4.9		.21	17	17		2.46	7.5	11.5		9.0			
ASPHALTINES %													12.7		
DISTILLATION													10.8		
IBP				458		378	472	474		317					
10%				598		479	538	504		657					
50%				697		731	598	557		800					
90%				760		Cracks	665	Cracks		Cracks					
EP				Cracks			718								
ASH WT. %	.045	.0045	.02	.0003	.15		.0003	.004	.0043	.0035		.009	.032	.016	.003
METALS PPM															
ASH	445	45	200	3	1500		3.5	14	43	30		93	320		
VANADIUM	55	7	7	.028	55	55	.005	.07	.33	1		11	95	24	.4
SODIUM	37	0	6	.125	12	9	.06	2.3	.11	1		4.0	7	6	.64
CALCIUM	0	0	30	0	335	0	.03	0	.28	4		1.1	9	5	
MAGNESIUM	0	0	0	0	0	194	0	0	.15	3		36.0	285	100	
LEAD	0	0	0	0	0	0	0	0	0	0			.52		
NICKEL		0		0	130	101	.005	.07	0	2			48		.95
IRON													16		
ADDITIVE	No		CA	No	CA	MG	No	No	No	No	No	MG		MG	No
BEGAN USE	7-1949		2-1952	5-1952	8-1952	12-1953	9-1953	3-1954	6-1954	6-1954	8-1954	8-1954			

TABLE I



TOP -- SIDE VIEW OF GAS TURBINE ELECTRIC LOCOMOTIVE SHOWING LAYOUT OF APPARATUS.

BOTTOM -- AIR FLOW DIAGRAM OF GAS TURBINE FOR LOCOMOTIVE DRIVE INSTALLED IN GAS TURBINE ELECTRIC LOCOMOTIVES CONSTRUCTED BY GENERAL ELECTRIC FOR UNION PACIFIC RAILROAD.

FIG. 5 LOCOMOTIVE APPLICATION OF SINGLE SHAFT OPEN CYCLE GAS TURBINE

FIG. 6 GAS TURBINE LOCOMOTIVE WITH INSIDE TURBINE AIR INTAKE

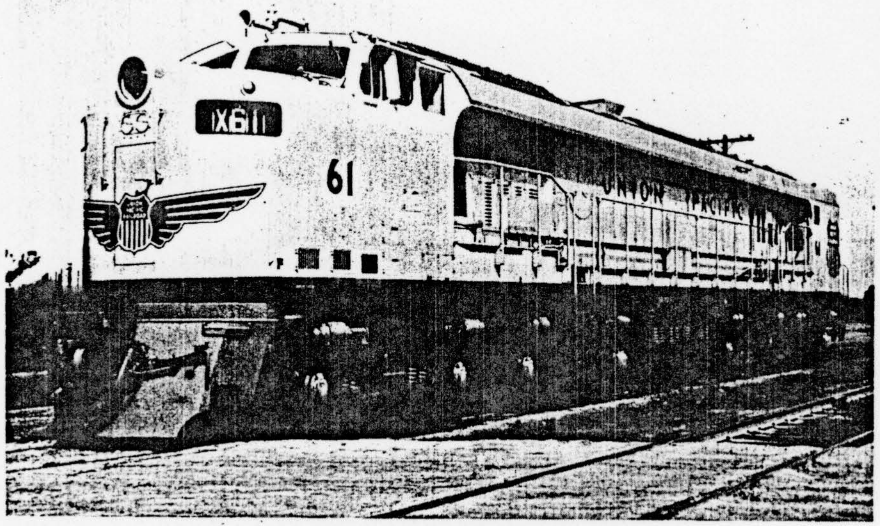


FIG. 7 GAS TURBINE LOCOMOTIVE WITH AIR INTAKE ON ROOF

FIG. 8 TURBINE WHEEL DAMAGED BY EXCESS TEMPERATURE

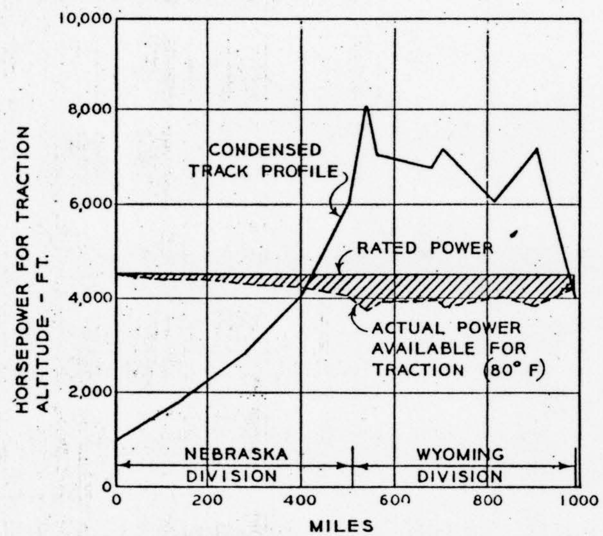


FIG. 9 GAS TURBINE POWER LOSS DUE TO ALTITUDE

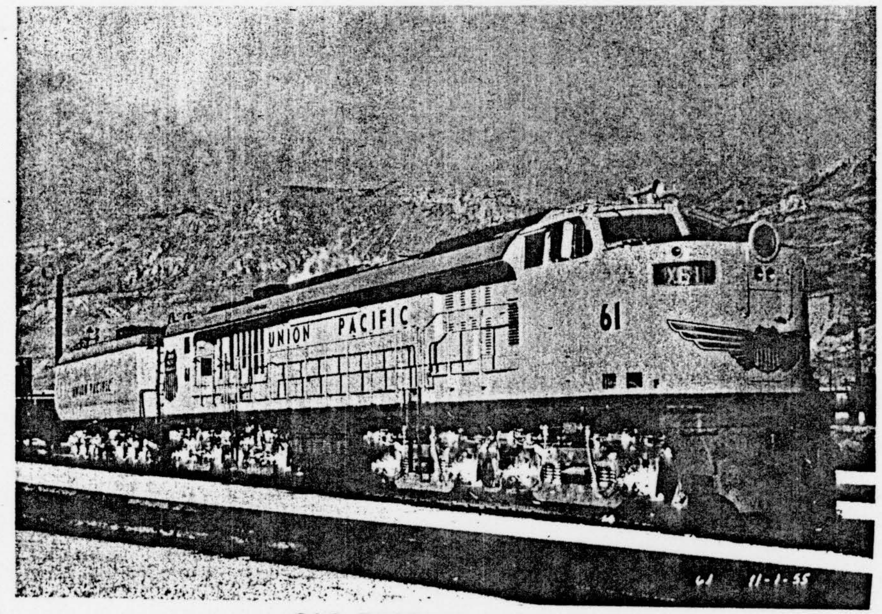


FIG. 10 GAS TURBINE LOCOMOTIVE WITH 24,000 GAL. FUEL TENDER