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DETACHABLE BLADE CONFIGURATIONS FOR A TOTAL FLOW GEOTHERMAL TURBINE

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ABSTRACT

The feasibility of using a simple tee-slot configuration to attach removable titanium turbine buckets to a geothermal turbine disk operating at 6000 RPM is considered. A finite element stress analysis was utilized and a possible design configuration with maximum tensile stresses of about 30,000 psi was found. Further reduction in this stress level could be made by optimizing the blade-disk geometry. No consideration was given to dynamic stresses caused by vibration.

INTRODUCTION

The first generation two-phase flow turbine constructed for the Total Flow Geothermal Energy Program incorporated non-detachable or integral buckets or blades on the turbine wheel. The wheel and bucket assembly was machined from a solid disk of forged aluminum using a numerically controlled milling machine. The method permitted the construction of a working turbine in a reasonable length of time which could be used to study the two-phase flow process and provide design data for improved turbines. Future generation turbines must be designed for testing in the actual geothermal fields and with consideration given to ultimate commercial use. Detachable, rather than integral, blading would seem a desirable feature of such turbines for a number of reasons. Erosion and/or corrosion may prove to be severe enough to necessitate occasional blade changes in which case the integral blade method may prove economically unsound. Detachable blades would facilitate experimentation with special fabrication methods for blades such as using coatings, plating and ceramic inserts. Detachable blades will also have much better structural damping properties which will greatly minimize the blade vibration problem. This report describes a study of the feasibility of designing a single wheel axial flow turbine having the same overall size as the existing first generation turbine at LLL, but having detachable blades.

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It was decided not to consider exotic or unusual design configurations which involve feasible but unproven or state-of-the-art technology, e.g., shrinking a fiber-wound composite band or shroud around the outer circumference of the blades and thus placing them in a very desirable state of compression or at least reduced tension. There was not enough time to prove out such design concepts. Preliminary consideration was given to many possible ways of fastening blades to the circumference of a disk, including most of the methods used past and present in the steam and gas turbine industries. Rough calculations quickly zeroed-in on the traditional method of keying the blade to the disk with some form of "Tee" slot in one part and a mating section in the other. (Figure 1.) The form of these slots can vary widely from a simple Tee shape (Fig. 1-a) to a complex "fir tree" shape (Fig. 1-c) chosen to minimize stress concentration. See Ref. 1, pp. 353-362 for further examples of blade attachment methods.

This report studies the Tee-slot-in-the-wheel configuration shown in Fig. 1-a and 1-b. Static stress analysis is performed using the finite element method with a number of approximations made as demanded by limitations of the computer codes used. Centrifugal inertia stresses are considered, but no dynamic stresses due to possible vibrations are considered. The study can be considered a feasibility study only to determine if the general design concept of detachable blades is possible.

Summary, Results and Conclusions

The "Tee-Slot" configuration for a detachable titanium blade shown in Fig. 2 is feasible with maximum tensile stresses less than 30,000 psi, which is less than the fatigue endurance limit of 40,000 psi.

Further reductions in stress levels should be possible if fillet design is optimized and especially if the fir tree root is adapted.

A split turbine disk utilizing these same blade types also is feasible. This disk design offers several possible advantages such as smaller forgings and easier machining.

Vibration studies of these blade and disk configurations should be carried out.

a

b

c

d

Fig. 1. Blade attachment methods.

Analysis Method

A. Interactive Computer-Aided Design

The problem was approached as an interactive-computer-aided design of a detachable wheel-blade configuration. Only that portion of the structure which included the turbine blade base, root and shank along with the turbine wheel rim region was thoroughly analyzed (Fig. 2). The rest of the turbine wheel including the central hub was ignored as far as detailed stress analysis was concerned. Stresses in the actual blade and in the region where the blade is attached to its own base were also ignored. Referring to Figure 2, the areas of prime concern were the blade shank and root and that portion of the wheel labeled rim region.

The general idea in using an interactive-computer-aided-design approach for such a problem is to take a proposed design, utilize a computerized stress-analysis code in an "on-line" or time-shared mode and quickly find out the stress field in the design when it is subjected to the service loads. The design engineer analyzes the results preferably on-the-spot, utilizes his engineering experience to alter the design to correct the problem of high stress levels and stress concentrations, enters these geometric design changes into the computer and repeats the process.

If the above method is to be really an interactive-computer-aided-design, there must really be a rapid man-computer interaction so that many design variations can be carried out. For stress analysis of mechanical parts having complex geometries, this requires:

1. A digital computer with excellent "number crunching" capability operating in a direct access or (more usually) a time shared mode. Batch processing is generally too slow.
2. A graphic display terminal.
3. Hard copy peripherals for immediate reproduction of both alphanumeric and graphic output.

4. Software in the form of a finite element stress analysis code or codes which are especially designed for use in an interactive mode. This software requires:
 - a. A fast way of "describing the part", that is entering the part geometry and properties into the computer. Repeated changes in part geometry should also be easy to make.
 - b. An automatic finite element mesh generator which requires minimal user input and is tolerant of design changes.
 - c. Immediate graphic display of the results in a form of a TV screen presentation of the undeformed and deformed part, mesh configuration, and stress and strain contours super-imposed on a display of the part.

The Lawrence Livermore Laboratory possesses Items 1, 2, and 3 above in their OCTOPUS in-house computer terminal network.

The software requirement (Item 4) was only partially satisfied for this project, but all things considered, a very reasonable compromise was found. This writer knows of no general purpose finite element stress analysis code which would completely satisfy the requirements listed under Item 4 above. The basic problem is input time. It is a tedious, time consuming process to input the geometry and properties of the parts and generate the mesh and any intelligent, capable engineer can quickly be "driven up the wall" by the difficulties encountered in getting the problem into the computer. Interactive design is a failure when most of the time is spent over such tiresome details and no engineer who wants to use the method as a design tool will, in fact, use it in the interactive mode if such difficulties are the case.

It is possible to write a special purpose code for a specific design situation which fulfills the requirements reasonably well. There was not sufficient time during this project to write such a code for turbine blade studies and it was necessary to use existing LLL stress analysis codes.

B. Stress Analysis Code DTVIS2

A precise stress analysis of the complex geometry found in a turbine wheel with detachable blades requires a three-dimensional finite-element

computer code. While such codes are available at LLL, they hardly fulfill the requirements listed above, especially given the fact that this user had only a limited time to familiarize himself with both the LLL OCTOPUS System and the LLL codes. The compromise was to use a two-dimensional finite element code called DTVIS2 capable of solving either axisymmetric or plane strain or plane stress problems. It was written by Dr. G. L. Goudreau, of LLL NEED Division, who also developed a mesh generator code (MESH2) and post-processor software (POST2) for output including TV graphic display of stresses, displacement, etc. Documentation for these codes is available from NEED.

This code proved dependable and met the condition for use in interactive design reasonably well.

C. Design and Operating Conditions

All calculations were based on a turbine operating at 6000 rpm with both the wheel and blade made of titanium alloy (Ti-6Al-4Va). Because of the low operating temperature of the turbine (54°C)(130°F), thermal stresses were ignored. No values of allowable stresses which would insure freedom from stress corrosion problems could be obtained, so a maximum allowable tensile working stress of 207 MPa (30 ksi) psi was chosen which is substantially below the endurance limit of about 276 MPa (40 ksi) psi for this material. Material properties used were (2, 3):

Alloy Ti-6Al-4Va

Density - $4.43 \times 10^3 \frac{\text{kg}}{\text{m}^3}$ (0.16 lbs/in³)

Young's Modulus E - 1.1×10^5 MPa (16×10^6 psi)

Poisson's Ratio $\nu = 0.3$

Yield Stress $S_y = 827.0$ MPa (120,000 lbs/in²)

Thermal Expansion Coefficient = 8.1×10^{-6} mm/mm-°C (4.5×10^{-6} in/in-°F)

All designs assumed a 102 mm (4-in.) high blade mounted on a 925 mm (36.4-in.) diameter wheel (i.e., diameter from blade center to blade center). Blade dimensions are given in Table 1 which is keyed to Figure 2.

D. Disk and Blade Stresses and Deformation

An accurate analysis of the stress-strain configuration of the blade attachment region at the outer circumference of the turbine wheel would require a three-dimensional finite element code. Such a code was not available, at least not in the time available. The spinning disk is a continuous structure and the attached blades are discrete sub-structures relatively free to slide with respect to each other and the slot.

An approximate method of analysis was worked out using both the axisymmetric and plane strain capabilities of DTVIS2.

Fig. 2. Blade dimensions.

TABLE 1
BLADE DIMENSIONS

(See Figure 2)

	"Dimensions in Inches"	
	2-inch Wide Blade	4-inch Wide Blade
d_1	2	4
d_2	0.862	1.72
d_3	1.75	4.2
d_4	0.25	0.25
d_5	0.5	1.5
d_6	1.0	1.5
d_7	0.5	0.5
d_8	0.25	0.5
R_2	0.050	0.050
R_3	0.25	0.25
R_1	0.85	1.225
Disk Width	2.6 to 3.0	4.0 to 4.6
No. of Blades	118	59
Blade Cross Sectional Area	0.375 in ²	1.5 in ²
Total Blade Inertia Force (Resisted by disk)	9074 lbs.	45,788 lbs.
Inertia Load on Blade Shank	7663 lbs.	36,223 lbs.

Analysis of the stresses and deformations near the rim of the wheel was a two-step process. First the centrifugal effects in the disk alone due to rotation of the disk at the design speed of 6000 rpm were considered by using the axisymmetric option of DTVIS2. No blades were attached to the disk. Only the portion of the disk near the rim was modeled in careful and correct geometric detail; the rest of the disk was treated as a solid spinning disk with uniform thickness (no central hole). A coarse finite element mesh was used for the inner portion of the disk and a fine mesh over the rim region. Because of geometric symmetry perpendicular to the axis of rotation, only one-half of the disk was modeled. (Figure 3.) The blades were not considered in this first portion of the analysis.

Next, the system of separate blades mounted in a tee-slot and subject to centrifugal loading was modeled by a plane strain/stress model. The plane strain option of DTVIS2 was the code used. Only the region near the rim of the wheel was modeled (Figure 4). The disk was treated as a plane strain problem which assumes a uniform stress perpendicular to the plane of the problem (R, Z plane). This simulated the axisymmetric nature of the disk problem which provides a uniform stress around the circumference for any value of R, Z.

The blade base, root and shank cannot be considered as a plane strain problem since the separate blades are free to move radially outward. The blades were considered to be in a plane stress condition and were so modeled in DTVIS2 by adjusting the materials properties of the blades to simulate a plane stress condition using the plane strain option of DTVIS2. This was accomplished by specifying two materials in the code. The regular titanium alloy (Ti-6Al-4Va) was used for the disk and a second material with plane stress adjusted values of the material properties was specified for the blade. Only the blade root base and shank was accurately modeled. A distributed load equivalent to the inertia load of the entire blade, base, shank, and root was applied to the extended shank (Fig. 4) to model the effect of the complete blade loading on the disk and to calculate the resulting stresses in the disk. A second run was made with a distributed load equivalent to the inertia load of the blade, blade base and shank (but not the root) to simulate the tensile loads in the shank of the blade caused by the rotating inertia load of these parts.

Fig. 3. Typical model of axisymmetric spinning disk.

Fig. 4. Plane strain/stress model of blade in tee slot.

The plane strain/stress model assumed a continuous or "welded" connection between the blade and disk at the surface of contact between the blade root and the under surface of the tee slot in the wheel rim.

Final answers for stresses in the disk were found by adding the stresses found for the axisymmetric spinning case and the plane strain/stress solution. The blade shank stresses were found directly from the plane strain/stress solution.

A further computation was necessary for blade configurations with a "lock" on the blade base such as is shown in Figures 2, 3, and 4. There was no direct way in the DTVIS2 code to simulate the condition that the two mating surfaces at the lock are in contact, but free to slide. Only the condition of a continuous or welded "joint" can be simulated. A thin interface element of low modulus material can be added but may lead to mesh generation problems. In this analysis, the mating face of the step in the disk was initially fixed as an exterior boundary with zero displacement in the Z direction and zero forces in the R direction (i.e., a "roller" boundary condition). The corresponding mating face on the blade base was left free and of course tended to bend away under load leaving a gap at the interface. After loading, the boundary forces existing "on the rollers" at the interface were provided by DTVIS2 and the "roller" boundary condition was removed. An equal and opposite external pressure distribution was placed on the two mating surfaces which were treated as external boundaries. The problem was run and checked to see if the displacements of the two surfaces agreed reasonably well (i.e., no severe gap or interference). About three iterations with adjustments to the applied external pressures usually were sufficient for a reasonable answer. (i.e., a negligible gap between the mating surfaces of the "lock".) Figure 18 shows the results of an incorrect pressure distribution on the lock surfaces causing an interference.

E. Results - Solid Disk

The basic blade configuration is shown in Figure 2. A typical computer model of the axisymmetric spinning disk is shown in Figure 3 which is a copy of the TV graphic output from DTVIS2 and POST2. A detail of the rim region

is shown in Figure 5 along with the finite element mesh generated by MESH2. The deformation of the disk due to the 6000 rpm spin alone is shown in Figure 6. Figures 7 through 10 show the stress contours for (respectively) the radial stress (σ_R), thickness stress (σ_Z), tangential stress (σ_T) and maximum shear stress τ_{max} .

A typical plane strain/stress model of the rim region alone showing the blade base, shank and root interacting with the tee slot in the disk rim is shown in Figure 4. A close-up of the root region and the finite element mesh is shown in Figure 11 and 12. Displacements are shown in Figure 13 where an incorrect pressure distribution on the lock surfaces is allowing large deformations. The stress contours are shown in Figures 14-17. Figure 18 shows the deformation before a correct pressure distribution has been applied to the mating surfaces of the lock to keep the lock surfaces in contact.

The stress contour plots provided quick identification of the areas of maximum tensile stress and a detailed print-out of the stresses acting on every element was available for an even more precise identification of the stress magnitude. As can be seen from Figure 7 and 14, the maximum tensile stresses occurred in or near the fillets of the tee slot in the disk and the root of the blade. Studies were made of a 4-inch wide blade having the same blade dimensions as the present turbine blade and a 2-inch wide blade with a geometrically similar configuration (Fig. 2 and Table 1). In each case, the width of the disk (see Figures 2, 3) was a major variable although other dimensions in the rim region were varied. Some results are given in Table 2.

Two-inch wide blades fastened to a 3-inch wide disk meet the design criteria of a maximum of about 30,000 psi tensile stress. The largest stresses are in the rim of the disk. The stresses in the blade root (21,000 psi) are well on the safe side. Even 4-wide blades could possibly be utilized if the allowable design stress of 30,000 psi could be revised upward to about 40,000 psi. Even lower stress levels for both blade widths can probably be achieved by changing the fillet dimensions, and for certain by changing to a "Christmas Tree" attachment scheme as shown in Figure 1-c.

TABLE 2

TWO-INCH WIDE BLADES
MAXIMUM TENSILE STRESS

Disk Width Inches	Stress Blade Root psi	Stress Disk Plane Strain psi	Stress Disk Centrifugal psi	Stress Disk Total psi
2.6	21,000	26,000	8,000	34,000
3.0	21,000	23,800	7,164	30,964

FOUR-INCH WIDE BLADES
MAXIMUM TENSILE STRESS

Disk Width Inches	Stress Blade Root psi	Stress Disk Plane Strain psi	Stress Disk Centrifugal psi	Stress Disk Total psi
4.0	38,000	39,000	~10,000	~49,000
4.4	38,000	34,000	~9,000	~43,000
4.6	38,000	32,000	8,500	40,500

Fig. 5. Axisymmetric spinning disk detail of rim region showing finite element mesh.

Fig. 6. Spinning disk - deformation due to spin alone.

- 1 = 2.7E+02
- 2 = 1.9E+03
- 3 = 3.5E+03
- 4 = 5.2E+03
- 5 = 6.8E+03
- 6 = 8.4E+03
- 7 = 1.0E+04
- 8 = 1.2E+04
- 9 = 1.3E+04

Fig. 7. Spinning disk radial stress σ_R (psi).

- 1 = -1.9E+03
- 2 = -1.4E+03
- 3 = -9.8E+02
- 4 = -5.3E+02
- 5 = -7.6E+01
- 6 = 3.8E+02
- 7 = 8.3E+02
- 8 = 1.3E+03
- 9 = 1.7E+03

Fig. 8. Spinning disk stress σ_z (psi).

1 = 7.0E+03
 2 = 7.9E+03
 3 = 8.8E+03
 4 = 9.6E+03
 5 = 1.1E+04
 6 = 1.1E+04
 7 = 1.2E+04
 8 = 1.3E+04
 9 = 1.4E+04

Fig. 9. Spinning disk tangential stress σ_T (psi).

Fig. 10. Spinning disk maximum shear stress $\tau_{\max}$ (psi).

Fig. 11. Plane strain/stress model root detail.

Fig. 12. Plane strain/stress model finite element mesh.

Fig. 13. Plane strain/stress model root deformation.

Fig. 14. Plane strain/stress model root region stress σ_R (psi)
 incorrect pressure applied to lock allows large displacements.

1 = -9.1E+03
 2 = -6.6E+03
 3 = -4.2E+03
 4 = -1.7E+03
 5 = 7.5E+02
 6 = 3.2E+03
 7 = 5.7E+03
 8 = 8.1E+03
 9 = 1.1E+04

Fig. 15. Plane strain/stress model root region stress σ_R (psi).

- 1 = -1.6E+04
- 2 = -1.2E+04
- 3 = -8.5E+03
- 4 = -4.8E+03
- 5 = -9.8E+02
- 6 = 2.8E+03
- 7 = 6.6E+03
- 8 = 1.0E+04
- 9 = 1.4E+04

Fig. 16. Plane strain/stress model root region stress σ_z (psi).

- 1 = 2.4E+03
- 2 = 4.7E+03
- 3 = 7.0E+03
- 4 = 9.4E+03
- 5 = 1.2E+04
- 6 = 1.4E+04
- 7 = 1.6E+04
- 8 = 1.9E+04
- 9 = 2.1E+04

Fig. 17. Plane strain/stress model root region stress $\tau_{\max}$ (psi).

Fig. 18. Plane strain/stress model incorrect pressure on lock surfaces.

F. Results - Split Disk

The concept of making the disk in two separate halves while still using detachable blades was also explored. Two versions of this design were explored. The first version is sketched in Figure 19. No attempt was made to accurately model the hub area which would of course be bolted together. The split disk concept would require thinner forgings; the "slot" area in the disk would be easier to machine; no blade insertion groove in the rim would be necessary along with an associated blade locking device and sealant material might be easier to introduce in such a design.

The stresses in the blade-disk root region and in clamping bolts located along axis A - A were studied. The same 2-inch wide blade specified above (Figure 2) was used, except that no lock tab was used.

One 0.25-inch diameter bolt per blade carrying a 1200 lb. force (~25,000 psi bolt stress) results in a tensile stress contour pattern shown in Figure 20. A 0.003 gap does open up but the maximum stress of 23,000 psi is substantially below the 40,000 psi design limit. The gap could be a serious problem from a stress corrosion standpoint, but the gap could probably be substantially reduced or eliminated by redesigning the rim region to allow heavier clamping bolts and pre-stressing. Locking tabs on the blade base would also improve the situation. The blade-disk configuration is clearly feasible.

A second version of the split disk design was considered with attention focused on the hub design. The disk-rim, blade attachment area was assumed to be the same as described above and the hub area as shown in Figure 21. This design permits a cantilivered turbine shaft. Stresses were in general well below the 30,000 psi maximum allowable working stress. See Figures 22-25.

Fig. 19. Split disk configuration.

Fig. 20. Split disk radial stress σ_R (psi).

Fig. 22. 3 inch mounting flange for slit disk radial stress σ_R (psi) in hub region.

Fig. 23. 3 inch mounting flange on split disk maximum shear stress $\tau_{\max}$ (psi) in hub region.

Fig. 24. 2.3 inch mounting flange for split disk
 radial stress σ_R (psi) in hub region.

Fig. 25. 2.3 inch mounting flange for split disk
 $\tau_{\max}$ (psi) in hub region.

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