May 30, 1972 N. DAMLIS 3,666,376

TURBINE BLADE DAMPER

Filed Jan. 5, 1971

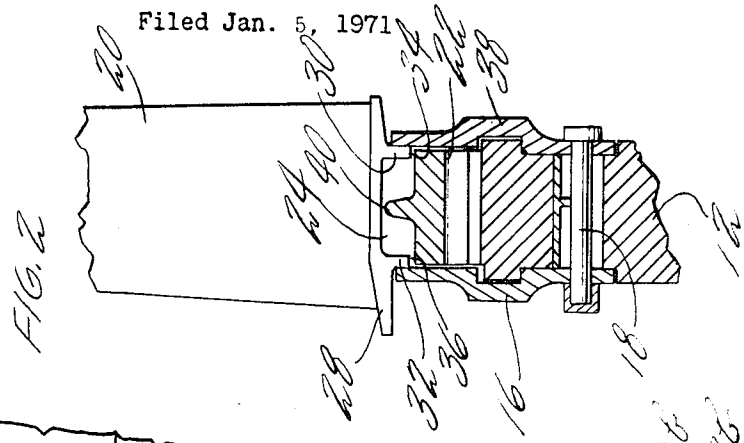


FIG. 2

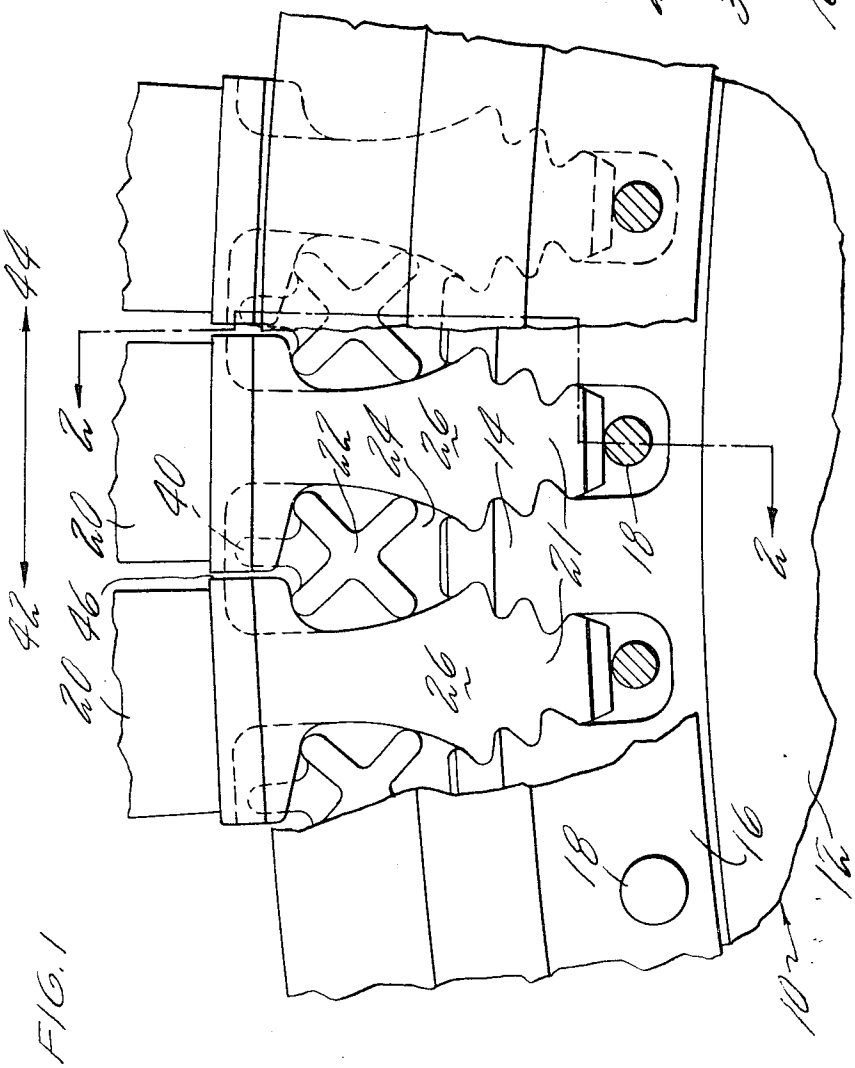


FIG. 1

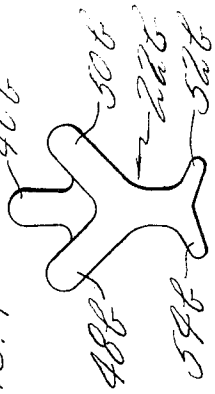


FIG. 4

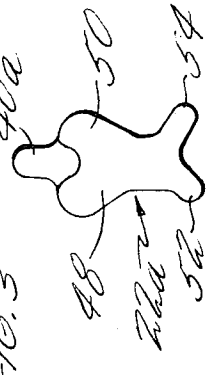


FIG. 3

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TURBINE BLADE DAMPER

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U.S. Cl. 416—219 5 Claims

ABSTRACT OF THE DISCLOSURE

A vibration damper for use in a turbine rotor of a high temperature gas turbine is disclosed. The damper which is X shaped in overall cross section with a pair of damping legs, a protruding assembly lug, and a pair of positioning legs is located in the turbine rotor in a cavity formed between the extended necks of a pair of adjacent turbine blades in one direction and end cover plates in the other direction.

BACKGROUND OF THE INVENTION

Field of the invention

This invention relates to gas turbine construction and more particularly to X-shaped vibration dampers used with gas turbine blades exposed to high temperatures.

Description of the prior art

The construction of blades for rotating equipment such as gas turbine or compressor wheels has always required precise engineering to assure that the blades are properly secured to the wheel and that an intolerable amount of vibration does not appear in the blades during operation. Turbine blade design is generally more critical than compressor blade design since turbine blades are exposed to hot gases and experience high stress levels which result from centrifugal, aerodynamic and vibratory loadings on the blades.

Generally speaking, turbine blades tend to change dimension under load in three different directions; axially—along the axis of rotation of the turbine wheel, radially—in the plane of the turbine wheel and tangentiallly—in the plane of the turbine wheel rotation.

The axial growth presents a relatively small concern because the blades can, for example, be held tightly by side plates attached to the turbine wheel. As the wheel expands and contracts, the side plates and the turbine blades experience corresponding dimension changes and no axial loosening of the blades occurs. Further, there is an axial clearance between adjacent turbine wheels and the change in axial dimension of a given wheel due to thermal growth does not result in any physical interference between adjacent turbine wheels.

Similarly, the radial growth presents a relatively small problem. Turbine blades are often attached to turbine wheels by the familiar fir tree attachment means wherein the contour of the root of the blade conforms closely to the contour of the mating surfaces in the wheel. As the wheel is rotated, centrifugal loading on the blade causes the blade root to maintain a positive contact at bearing surfaces common to the blade and the wheel.

The tangential growth is more difficult to accommodate. Unless an appropriate room temperature clearance is left between the platforms of adjacent turbine blades, the blades will be physically restrained from expanding in the tangential direction when they are heated sufficiently. Jamming or mechanical interlocking of the platforms of adjacent turbine blades and the consequential high stress levels can be eliminated by providing tangential clearance between adjacent blades, however, the blades then tend to vibrate during operation.

Tangential vibrations are sometimes controlled by constructing tip shrouds on the blades. The shrouds connect the tip sections of the adjacent blades and tend to damp and make more consistent the tangential vibrations which occur in the blades. As the gas turbine engine performance has improved, the temperature to which the turbine blades are exposed has increased and the shroud concept has become less attractive because cracks occur in the shroud section of the blade and propagate downward through the blade, resulting in premature failure of the blade. Further, turbine blade shrouding is conceptually undesirable because the diameter of the section of the engine where the shrouding is required tends to increase; also, the shrouding results in a high concentration of mass at a location which is relatively distant from the centerline of rotation, resulting in structural limitations in the engine. Therefore, a trend towards shroudless turbine blades exists. Since the shroudless blade is particularly susceptible to tangential vibration in the airfoil section of the blade, various devices have been resorted to in an attempt to minimize such vibrations. One of the accepted vibration damping schemes is a toggle device. A toggle is a nonstructural member which is held in physical contact with a turbine blade by centrifugal force during rotation of the turbine engine. The toggle reduces the vibratory action of the blade by imposing a retarding force due to friction on the blade at the point of contact between the toggle and the blade. When turbine blades are damped by toggles, each blade tends to vibrate at a frequency which is independent of that in the adjacent blades and the resulting stress levels in some blades becomes high with respect to that in other blades in the same wheel. Also, toggles impose relatively high centrifugal loads on their retaining structure. Further, space limitations impose restrictions on the mass of a toggle which in turn limits the damping force imposed on the blades. Therefore, engine manufacturers are continually searching for alternate inexpensive vibration dampers for use in high temperature, gas turbine applications.

SUMMARY OF THE INVENTION

An object of the present invention is to provide damping of the tangential vibrations in turbine blades which can cause relatively high stress levels in the blades and alternately structural failure in the blades.

According to the present invention, a high temperature gas turbine vibration damper having an X-shaped cross section is located within a turbine rotor between the extended necks of a pair of adjacent turbine blades which are connected to the wheel of the rotor, the damper being pressed against platform sections of the blades by centrifugal force caused by rotation of the turbine rotor, the centrifugal force resulting in a damping friction force between the damper and the blades which dampens tangential vibrations otherwise occurring in the turbine blades during operation. The X damper is free to move within a cavity bounded in the tangential direction by the extended necks of a pair of adjacent turbine blades and in the axial direction by side-retaining plates which contain the turbine blades in a sandwich fashion. The damper (1) is positioned to exert a blade damping friction force during rotation of the turbine rotor, (2) is shaped to interact with the walls of the cavity and avoid rotation about its own longitudinal axis, and (3) is free to move radially and avoid binding of adjacent turbine blades during thermal cycling and shutdown of the engine.

One feature of the present invention is that misorientation of the vibration damper is not possible due to the unique physical design of the damper. In addition, the damper is manufactured completely by a casting process which requires no machining and results in reduced production costs. Also, the X shape produces a tailored damp-

ing force which can be greater than can be produced by alternate damping devices because the X damper allows a greater mass to be installed in any particular extended neck blade assembly. Since a turbine blade having high vibratory stress is damped with a concomitant increase in the vibratory stress in its neighboring blades by the action of an X damper, there is a more even distribution of vibratory stress throughout all the blades in a turbine rotor; the blades which tend to experience high vibratory stress are damped and the blades which tend to experience lower vibratory stress are excited, with all blades on a wheel approaching a common intermediate stress. Further, since sufficient clearance is provided between the platforms of adjacent blades, binding of adjacent blades is avoided although the damper is free to move radially under changing conditions of rotational speed and temperature.

The foregoing and other objects, features and advantages of the present invention will become more apparent in the light of the following detailed description of preferred embodiments thereof, as illustrated in the accompanying drawing.

BRIEF DESCRIPTION OF THE DRAWING

FIG. 1 is an elevation view of a portion of a turbine rotor broken away to show the X-shaped damper in accordance with the present invention.

FIG. 2 is an auxiliary view taken along the lines 2—2 in FIG. 1.

FIG. 3 is a front elevation view of an alternate embodiment of an X-shaped damper in accordance with the present invention.

FIG. 4 is a front elevation view of another alternate embodiment of an X-shaped damper in accordance with the present invention.

DESCRIPTION OF THE PREFERRED EMBODIMENT

A portion of a turbine rotor assembly **10** is shown in FIG. 1. A turbine wheel **12** having fir-tree type connectors **14** on the periphery and a cover plate **16**, held to the turbine wheel by rivets **18**, is intimately connected with turbine blades **20** at root sections **21**. An X-shaped damper **22** is placed in a noncircular cavity **24** which is formed between the extended neck sections **26** of a pair of adjacent turbine blades.

A view through the blade attachment and the X-shaped damper is shown in FIG. 2. The turbine blade **20** has a platform section **28** and a pair of radially extending platform ribs **30** and **32**. The damper **22** which contacts the turbine blade along the top surfaces **34** and **36** of the ribs **30** and **32** respectively, is contained between the front cover plate **16** and a rear cover plate **38**. The X damper has an assembly lug **40** which extends in the radial direction between the ribs **30** and **32** in the cavity **24**; the lug prevents misorientation of the damper during rotor assembly since it is impossible to fit the damper into the cavity **24** with the lug pointed in any direction other than toward the platform **28**.

During rotation of the turbine rotor under operating conditions, the turbine blades will experience tangential vibrations in the direction **42–44** shown in FIG. 1. Adjacent turbine blades have a tangential clearance **46** at room temperature between their respective platform sections to allow for tangential growth during thermal cycling.

Rotation of the turbine rotor causes the damper to be forced radially outward, away from the center of the engine against the ribs of the turbine blades. For example, refer to FIG. 3 which shows an alternate embodiment X damper. A damper **22***a* has a pair of damping legs **48** and **50**, an assembly lug **40***a*, and a pair of positioning legs **52** and **54**. When the turbine rotor is rotating, the damping leg **48** bears against the surfaces **34** and **36** of the ribs **30** and **32**; the damping leg **50** would also bear against a corresponding pair of surfaces on the adjacent turbine blade.

Each of the damping legs **48** and **50** must rest on a different blade in order to transmit a damping force and thereby reduce the maximum vibratory stresses experienced in the blades. If a damper were to contact only one blade, vibration of that blade would cause the damper and blade to move as a unit; unless some relative motion occurs between a blade and a corresponding damper leg, no vibration retarding friction load is imposed on that blade. Alternatively, if a damper is in contact with a pair of adjacent blades, vibration of one blade of the pair is resisted by the rubbing action imposed thereon by the damper which is also being acted on by the other blade of the pair. Those familiar with the art should recognize that X dampers as described herein tend to force all the blades on a turbine rotor to respond as a single system, with those blades which tend to vibrate at relatively high frequencies becoming deexcited by frictional damping loads and also by increasing the vibration rate of neighboring but lower frequency neighboring blades. The result of this phenomenon is that the vibratory stress experienced by each of the individual turbine blades in a given rotor more closely approaches some uniform value which applies to substantially all the blades on that rotor; since all the blades are responsive to the vibrations of their neighboring blades, any one blade which tends to vibrate at a relatively high frequency will be damped and tend to excite its neighbors.

Each blade experiences a greater damping force under the action of an X damper when the damper bears against two blades simultaneously than would otherwise be possible with the maximum size toggle damper which might alternatively have been used, since a toggle generally bears against one turbine blade and also a rigid portion of the turbine rotor. Such a toggle would transmit one half of the centrifugal force it experienced to the blade and one half to the rotor.

The damping effect caused by an X-shaped damper results from the friction load which the damper imposes upon the blade as the blade vibrates in a tangential direction while the damper is forced against the ribs of the blade in the platform section due to the rotation of the turbine rotor. When any given turbine blade vibrates, relative motion occurs between the blade and the damper; at the points of contact between the damper and the turbine blade, friction forces are experienced due to centrifugal force on the damper and the force opposes the tangential motion of the blade. Therefore, the tangential vibration of the blade is decreased by the damping action or friction loading which the damper imposes upon the blade.

The friction loading can be calculated to be the product of the coefficient of friction between the damper and the blade and the force (centrifugal) normal to the top surfaces of the ribs at the point of contact. The coefficient of friction is a function of the materials and the condition of their surface in any given instance and there is little opportunity to substantially affect the damping force by varying the coefficient of friction. On the other hand, the normal force is more easily varied in order to customize the damping force on a turbine blade. The normal force is effectively the centrifugal loading of the damper due to the rotation of the turbine rotor and can be described by the equation

$$N = WRV^2$$

where

N is the force normal to the surface,
W equals the weight of the damper,
R is the distance between the center of rotation of the turbine rotor and the center of gravity of the damper, and
V is the rotational speed of the damper.

For a given turbine speed, a change in the weight of the damper has a significant effect upon the force that the damper can impose on a vibrating turbine blade.

Another alternative embodiment of an X damper is shown in FIG. 4. The positioning legs **52***b* and **54***b* intersect with the extended neck sections of the blades to prevent the damper from rotating about its own axis and assuming a balanced position so that only one damping leg is resting against a turbine blade. If a given damper were to contact only one turbine blade, no damping action would occur and the uninhibited blade could vibrate sufficiently to cause structural damage. The X-shaped damper is designed with enough mass to limit the tangential vibration of the blades so that the vibratory stresses are maintained within tolerable ranges. The damper is massive enough to provide the required friction loading upon the blade, and it is also free to move within the cavity and avoid jamming during thermal cycling. The damper cannot, however, rotate due to the design of the positioning legs and the fact that the cavity is noncircular.

The X damper is supported only at its ends, any given damper being in contact with a given turbine blade at but two points and any given damper having a total of four turbine blade contact points. Since the function of platform ribs is to support the blade platform, there is no need to extend the rib across the entire platform span of the blade; this is a weight-saving feature and is not critical to the invention.

Although the invention has been shown and described with respect to preferred embodiments thereof, it should be understood to those skilled in the art that the foregoing and other changes in the form and detail thereof can be made therein without departing from the spirit and scope of the invention.

Having thus described typical embodiments of my invention, that which I claim as new and desire to secure by Letters Patent of the United States is:

1. In a turbine rotor having an axis of rotation, a turbine wheel with a plurality of turbine blade root connection means on the periphery of the wheel, and a plurality of turbine blades which are attached to the wheel, one at each of the connection means, with each blade having an airfoil section, a platform section, a root section and an extended neck section which is intermediate of the root and platform sections, with adjacent blades forming a noncircular cavity between the extended neck sections thereof, blade vibratory damping means comprising:

a damper having a longitudinal centerline and a plurality of legs extending radially out from said centerline, the damper being fitted into the noncircular cavity with the longitudinal centerline of the damper being parallel to the axis of rotation of the rotor, each damper having at least a pair of damping legs each of which engages the platform section of a respective one of the blades forming the cavity to impose a friction force to the blade during rotation of the rotor, and each damper also having a pair of positioning legs to engage the walls of the cavity and prevent rotation of the damper about its longitudinal axis within the cavity.

2. The damper according to claim **1** wherein an assembly lug is located intermediate of the damping legs to prevent misorientation of the damper in the cavity.

3. A turbine rotor comprising:

a turbine wheel with turbine blade connector means;

turbine blades with each blade attached to the wheel and having an airfoil, a platform, and a root section, the platform being intermediate of the airfoil and the root section and having support ribs which extend from the root section, the root having an extended neck section and being shaped to conform to the blade connector means on the wheel;

end plates which are attached to the surfaces of the turbine wheel and which sandwich both the roots of the blades and the blade connector means; and

dampers, with each damper having an overall X-shaped cross section consisting of a pair of positioning legs, a pair of damping legs and an assembly lug which is located intermediate of the damping legs, the dampers fitting into a cavity formed by the extended neck sections of each pair of adjacent blades, and contacting the ribs of the blades during rotation of the rotor due to centrifugal loading on the dampers.

4. A turbine blade damper which is X-shaped in cross section and which is fitted into a noncircular cavity formed in a turbine rotor between a pair of turbine blades having extended neck root sections, the damper having a pair of adjacent positioning legs to engage the walls of the cavity to prevent rotation of the damper within the cavity, a pair of adjacent damping legs to engage the pair of adjacent turbine blades which form the cavity to damp vibrations in the blades and an assembly lug which is located between the damping legs.

5. A damper of X-shaped cross section which has a pair of damping legs to engage a pair of turbine blades which are adjacent each other in a turbine rotor to dampen vibration within the blades, a pair of positioning legs which cooperate with the extended neck root surfaces of the pair of adjacent blades to prevent turning of the damper about its longitudinal axis, and an assembly lug which is located between the damping legs to prevent misorientation of the damper.

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