Aerodynamic study for the flow through cascade blades of gas turbine with vibration effect: a review

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Abstract. A research survey on aerodynamic with/without vibration effect and investigations for turbine components of gas turbine engines is presented. Experimental and numerically predicted results are presented from investigations undertaken over the past 70 years. The aerodynamics between the turbine blades is very important as it determines the overall system performance. The importance of aerodynamics is being in the pressure distribution above and below each airfoil, the locations of the vortices occurring, their number, distribution, direction, and shedding, and the determination of the regions of fluid separation from the wall that is directly related to the fluid velocity. Many ranges of vibration that affect different bodies were taken, and some of these ranges were small (10-200 Hz), and others had large ranges from 1.5 to 10 KHz. Therefore, the result evolution and shedding of vortices corresponding to the deformation in fluid structure depends on the Reynolds number and value of amplitude and frequency for vibration. The Mach number is a clear indicator to represent these characteristics. It also includes studying the theoretical methods used and simulations of the researchers' aerodynamics, the accuracy of the results reached, the representation of the body contour and the diagrams for each of the pressure and Mach number, the pressure (\(C_p\)) and lift (\(C_L\)) coefficients, and their amount of change occurring at each location of the airfoil. The importance of the above topics has motivated us to prepare a comprehensive review mostly of the aerodynamics of turbine blades and the effect of the vibration on the flow. The main concepts of aerodynamics, such as \(C_p\) and \(C_L\) as well as vibration frequency are presented. It also highlights the performance of the turbine blades as a result of vibration

Keywords: aerodynamic, Flow, Cascade Blades, Gas Turbine, Vibration Effect.

1. INTRODUCTION

1.1. Importance of Gas Turbine
Gas turbines have continuously increased usage in primary and secondary aerospace, aircraft structures, and electric power generation owing to their superior output power and high mechanical torque [1]–[5]. Due to the wide range of applications of the gas turbine, its aerodynamic characteristics have attracted many researchers and have been considerably improved to achieve realistic results. The aerodynamics of the flow in a turbine stage (stator/rotor) is rather complex and is still the subject of
many ongoing research activities in the gas turbine community[6]. The flow is inherently three dimensional (3D) due to the vane/blade passage geometry with features such as twisting of the vane/blade along the span, clearance between the blade tip and the shroud, film cooling holes, and end wall contouring [7]. The passage flow is characterized by boundary layer effects, secondary flows generated by the passage pressure gradients, and vertical flow structures such as the leading edge horse-shoe vortices, tip-leakage flow vortices, and corner vortices[8]. At the higher flow rates, the flow becomes choked. The stall is caused by large flow separation at the suction side of the blade profile which occurs when the fluid incidence angle is too high or too low. It can lead to severe blade vibration and, as consequence, machine failure. As a result, investigations have been working for many years to better understand of the aerodynamic flow through gas turbine cascade blades with vibration effect[9], [10].

![Figure 1](image1.png)

**Figure 1.** Streamlines showing separation lines in a near end wall plane of a linear blade passage (Lakshminarayana 1996).

1.2. General observation of some early past investigations

In the last seven decades, flow visualization techniques were employed by Herzig et al. [11] that directly identified the existence of vortices in cascade passages. Even before this time, other experimental and theoretical studies attempted to give a qualitative understanding of secondary flow behavior and to provide meaningful approximations of loss calculations [11]–[13]. In 1955, Hawthorne [14] introduced the secondary flow vortex system shown in Figure 2.
In this model, Hawthorne suggests that flow passing through a cascade may be divided into three parts. The first is a distributed secondary circulation that is the result of a distortion of vortex filaments carried. The first is a distributed secondary circulation that is the result of a distortion of vortex filaments carried with the stream and arises inflow around bends and between cascade passages. The second part is a trailing shed vortex due to the change of circulation about the airfoil in the spanwise direction[15]. The third component is trailing filament vortices, which arise due to the vortex filament on the convex surface of the blade moving ahead of that on the concave surface, thus causing it to
stretch. Hawthorne also noted that the stretched filaments contribute to the vorticity of the same sign as the trailing shed circulation, and that, together, they form a trailing vortex sheet which separates the upper and lower parts of the flow from each other[10].

The flow field near the hub endwall region of the blade passage is dominated by the boundary layer, strong pressure gradients, and cross-flow in the pitchwise direction from the pressure side to the suction side[16]–[18]. The resulting near-wall flow field is complex and consists of strong secondary flows and vortex roll-up [19]. When the endwall boundary layer approaches the blade row, a vortex is formed near the junction of the blade leading edge and the endwall. This vortex is termed as the leading edge horseshoe vortex. The horse-shoe vortex splits at the leading edge and propagates downstream into the passage on both the pressure side and the suction side of the blade passage forming two legs of the early passage vortex flows[20]. Corner vortices are also induced in the corner formed by the blade and the hub endwall.

Figure 3. Streamlines and static pressure distribution in the mid-span plane along blade passage.

The streamlines slightly above the endwall in Figure 1 show some distinct features of the endwall boundary layer flow. These features are identified by the separation lines in the figure. The streamlines along the blade leading edge bifurcate as they approach the saddle point. The saddle point is the location on the endwall where the zero degree incidence line meets the separation line and corresponds to the lowest friction velocity. The incoming endwall boundary layer detaches along the separation line, and secondary vertical flows are formed in the regions immediately downstream and adjacent to the separation line. This was indicated by the high concentration of the streamlines adjacent to the separation line. The strong reverse flow in the vortex regions counter the boundary layer streamlines causing them to be concentrated more densely near the separation line. The leading edge horse-shoe vortex immediately downstream of the saddle point is clearly evident in Figure 1. The region between the separation line and the blade suction side in figure 1 represents the suction-side leg of the horse shoe vortex [19]. The region along the separation line directed from the pressure side to the suction side represents the pressure-side leg of the horse-shoe vortex [19] and is driven by the passage pressure gradients. The suction side leg vortex and pressure side leg vortex meet together in the mid-passage region where the two separate lines in the passage merge. This location occurs close to the suction surface, and the merger of the two vortices forms a stronger vortex known as the passage vortex. The passage vortex then travels along the blade suction surface toward the passage exit. The endwall boundary layer region is very thin and skewed toward the suction side. This is evidenced by the streamline concentration being sparse in this region as they turn from the pressure side to the suction side. The strong vertical motions of the pressure side leg vortex entrain most of the fluid from
the incoming boundary layer and a new boundary layer forms downstream. Comparing the streamlines in Figure.1 with those in the mid-span regions in Figure.3, it is clear that the turning of the streamlines inside the blade passage and around the leading edge is much greater near the endwall region which causes the cross-flow here to be stronger.

The flow yaw angle contours along the passage in Figure.4 show the higher magnitudes of the flow turning near the endwall compared to the flow turning in the mid-span regions away from the endwall. The endwall pressure gradients are shown in Figure.5. The suction side pressure magnitudes along the endwall are higher compared to the suction side pressure values in the free stream region (see Figure 1). This results in smaller pressure gradients in the endwall from the pressure side to the suction side as shown in the line plot of Figure.5. The magnitude of $\Delta P$ in the Figure is the pitch wise pressure difference between two points at the same axial chord location, one located on the pressure side and the other on the suction side. As mentioned earlier, the turning of the boundary layer fluid in the endwall region is much higher compared to the turning of the free-stream in the passage, which seems to contradict the results in Figure.5. According to Niehuis, et al., in the free-stream flows away from the endwall, equilibrium exists between the pitchwise pressure gradient and centrifugal force on the fluid elements at the curved streamlines [21]. This equilibrium breaks down in the endwall region because the centrifugal force on the fluid elements in the low-velocity boundary layer reduces. As such, the weak endwall region streamlines easily turn to a greater degree with relatively smaller pressure gradients than those in Figure.5.

Based on these observations and for more convenience, the previous works have been classified based on the study kind in the following sections.

1.3. Theoretical Studies

Whitehead [22] presented a theoretical method of calculating the aerodynamic forces on vibrating compressor and turbine blades subject to two fundamental limitations. These were, firstly, that the time for the fluid to flow through the cascade must be small compared with the period of the vibration,
and secondly, that adjacent blades must vibrate nearly in phase. This enabled any kind of two-dimensional steady-flow cascade data to be used. The methods are given also allowed the vibration forced by fluctuations in the incident flow to be calculated. In that paper, the vibration on the flow stream and vorticity were governed by the equations.

\[
\lambda = \frac{\omega c}{U\alpha} \ll 1
\]  

(1)

\[
\psi = Ae^{(c_x-iy)\omega/v_s}
\]  

(2)

\[
\zeta = \nabla^2 \psi
\]  

(3)

With regards to aerodynamic coefficients, the pressure, force and moment coefficients were presented as

\[
\psi_z = \frac{p_2 - p_1}{\frac{1}{2} \rho v_s^2}
\]  

(4)

The researcher did not present a general flow visualization between the form feathers of the body contour. No clear representation of the composition, number of vortices, and its shapes. However, a clear distinction about the factors \(C_p\), \(C_L\), \(C_M\), and \(C_F\) was presented. The mathematical representation of the model involved all the parameters that consider the vibration effect. The effect of vibration on the performance of the blade was given in the details.

Whereas Gerolymos in 1988 [23] developed an algorithm for numerically integrating the unsteady Euler equations in blade-to-blade surface formulation. The method simulated all the inter-blade channels (passages) of an annular cascade. He divided the passage into regions. The equations were discretized in a grid that moves in order to follow the vibration of the blades. The equations were integrated using the explicit MacCormack scheme in the finite-difference formulation. Mistuned vibration as well as standing-, traveling-, or influence-wave modes may be readily simulated. Also, two other faster methods were developed, simulating traveling and influence waves, respectively. A number of numerical results showed the aptitude of the method to simulate both started and unstarted supersonic flow in vibrating cascades. A first comparison with available wind-tunnel data corroborates the validity of the approach to predict supersonic flutter of fans and compressors.

In that paper, the vibration frequency on the flow was set as a constant value. A good presentation about the pressure diagram in the suction and pressure area. Mach number distribution was clearly presented for understanding how shock waves could occur. In addition, converged results were presented to validate the accuracy of the results. However, the researcher could not clearly present the pressure and velocity of the fluid around the wing in the form of body contours. No clear descriptions of the developed vortices were presented.

Moffatt et al. [24] presented a decoupled aeromechanical system based on an advanced frequency-domain CFD solver with fillly non-linear capability allowing resonant vibration predictions to be routinely performed during the design process. A new energy method was presented that solves the blade response without the knowledge of original mode shape scales. A robust FE-CFD mesh interface was developed for industrial use that can accurately deal with differences in mesh geometry, low mesh density, and high mode shape gradients. The capability of the baseline CFD solver for blade row
interaction flow prediction was further validated against the VKI transonic turbine stage. A forced response analysis was carried out on the NASA Rotor 67 transonic fan for demonstration purposes. The system was evaluated for a challenging industrial study of the ALSTOM 3-stage transonic test compressor. Where the forced response predictions of three crossing points on the Campbell diagram compare well with strain gauge test data. An investigation into aerodynamic damping of the first ten modes shows a high dependency on mode shape. In general, the authors did not mention the N-S equations in clear form and the hypothesis required for that. They only focused on the appearance of the shock waves and did not focused on the importance of the reassurance distributing pressure and creating vortices further. In addition, they did not take into account the theory and method of calculating vortices such as K-ω and k-α. However, they properly used the finite element method for solving the vibration effect on the blades, correlated them with aerodynamics in a new way, and then calculated the resulting deformation. In addition, high levels of vibration were studied to give a clear visualization of what is happening at those levels and their effect on regular design.

Forbes and Randall [25] presented an analytical model of the casing and simulated pressure signal associated with the rotor blades in order to understand the complex relationship between blade vibrations and casing response. A mathematical formulation was undertaken of the internal pressure signal due to both the rotating bladed disk as well as individual blade vibrations and the solution of the casing response was formulated. Excitation by the stator blades and their contribution to the casing response was also investigated. Some verification of the presented analytical model was provided by comparison with Finite Element Analysis results for various rotor rotational speeds. The forces acting on the blade are therefore modeled as a raised cosine with a period of half the blade pass forcing frequency. That is resulting from the case study was on solid (tip of blades and casing).

\[
f(t) = F_0 \sum_{i=0}^{\infty} A_i \cos \left\{ i \left( \omega_{sp} t + \gamma_r \right) \right\}
\]

(5)

\[
\gamma_r = \frac{2\pi(r-1)}{b} - \text{round} \left\{ \frac{s(r-1)}{b} \right\} \frac{2\pi}{s}
\]

(6)

Where \( A_i \) are the Fourier coefficients and \( \omega_{sp} \) is the stator passing frequency.

The equations

\[
P_n = P_e \left( \theta - \Omega t - \chi(t) \right) - H \left( \theta - \Omega t + \frac{\pi}{b} \right) - H \left( \theta - \Omega t - \alpha_r + \frac{\pi}{b} \right)
\]

(7)

Gareth L. Forbes and Randall [26] introduced an analytical model of a gas turbine casing and simulated pressure signal associated with the rotating blades, individual blade vibrations, and transfer of stator blade vibrations in order to understand the complex relationship between casing response and the most important excitation forces. Due to the force interaction being through a fluid medium, a certain degree of randomness was introduced into the excitations, and the viability of this inherent randomness as a useful aid for separation of the contributing excitation forces from the system response was explored analytical solution for a circular ring, under the simulated operating conditions for a gas turbine with both periodic and turbulent force components included, has been presented. Importantly, the random contribution to the excitation of the casing and the rotor blades has been shown to contain information about the blade vibration characteristics. The separation of the different force signal types was also shown, with periodic contributions removed and the potential ability to separate the contribution to the casing vibration for different stages by taking advantage of the signal’s inherent cyclostationary properties.
In this work, the vibration frequency, and pressure profile were governed by the equations. The forces acting on the rotor blades are therefore modeled as a raised cosine with a period of half the stator blade pass frequency, and modulated by a random fluctuation due to turbulent flows. The Fourier series expansion of this force on the \( r \) th blade can be expressed as

\[
f(t)_{r1} = F_0(b(t) + 1) \left( \sum_{i=0}^{\infty} A_i \cos \left[ i(\omega_{sp} t + \gamma_r) \right] \right)
\]

(8)

\[
P_r = \sum_{i=0}^{\infty} A_i \cdot P_e \cdot e^{i[\theta - \Omega t - x(t) - \alpha_r]}
\]

(9)

The rotor blade motion such that the resulting moment for the \( P \) th, Stator blade can be expressed as:

\[
T_p = M_0 \left( \sum_{r=1}^{b} \sum_{i=0}^{\infty} A_i \cos \left[ i(\Omega t - x(t) - \alpha_r - \alpha_p) \frac{\pi}{b} \right] \right) \frac{\delta(\theta - \alpha_p)}{R}
\]

(10)

where \( A_i \) is the Fourier coefficients, \( \omega \) is the stator passing frequency, \( b(t) \) is a uniformly distributed random variable with zero mean and a deviation of \( \pm 7.5\% \). With regard to Equation (5), the researchers did not take the effect of random vibration that could be added to the vibration applied to the blade in Equation (8). Therefore, they reformulated Equation (5) to take this aspect into consideration, as in Equation (8). The authors did not take into consideration the variable vibration ranges with the constant amplitude of the frequency since the distance between the tip of the blade and the case of the turbine is a fixed design distance, where the researcher used different amplitude for each vibration. Next, the effect of vibration on the air aerodynamics in that region and the possibility of taking advantage of the characteristics of the dynamic or avoiding it as it gives importance to the performance of the system. However, the beginnings of the vibration were wide and gave a visualization of every range of vibration being between \( (0 – 2000) \) Hz and the response resulting from this vibration. In addition, the possibility of adopting this research in the observation of the performance of the organization and the manner of the response resulting from the vibration and knowledge of the ranges of operation, and stages of operation risks, which must then be shut down the system.

In his work, Khan and Wei, [27] carried out a CFD analysis to study the influence of local flexible structure on the lift performance of airfoil. Single-mode vibrations of local flexible structure on the upper surface of the airfoil were investigated with various frequencies and amplitudes. Finite Volume Method was used to discretize the incompressible Navier-Stokes equations. Partial fluid-structure interaction was examined under the dynamic mesh technique. The flow configurations were assumed to consist of flow over NACA0012 at \( Re=1000 \) and at a low angle of attack. Numerical simulations were conducted for different vibration frequencies and the corresponding flow field characteristics were investigated as well. Vibrations with a range of amplitudes were carried out under the same flow conditions and their results were compared with the conventional rigid airfoil. Vibration frequency close to the natural vortex shedding frequency resulted in the frequency lock-in or synchronization phenomenon. At the frequency lock-in condition and at some moderate vibration amplitude, the amplitude of lift coefficient oscillations was increased. These vibrations on the upper surface of the airfoil resulted in increasing the average lift coefficient, which was further increased with increasing the vibration amplitude. In addition, the displacement of the local flexible structure was completely in
phase with the vortex shedding. The length of the local flexible structure was also varied between 0.1c and 0.2c and results showed that the length of 0.1c had better performance at higher vibration amplitudes. Pressure distributions and vortices around the local flexible airfoil were evaluated. Moreover; the power spectra based on the lift coefficient were tested to imprisonment the flow field energy. The effects of vibration frequency (fv) are studied at the constant vibration amplitude (Av) of 0.006c. The vibrations are modeled as y(x, t), where y is the position of a point x on the LFS at time t. The governing equation for vibrations of LFS is;

\[
y(x, t) = A_v \sin \left( \frac{x - x_2}{x_1 - x_2} \right) F(t)
\]

Khan et al. [28] studied partial fluid-structure interaction of an airfoil. The 2D incompressible Navier-Stokes equations were solved based on the finite volume method and dynamic mesh technique. The local flexible structure was assumed to vibrate in single-mode located on the upper surface of the airfoil. The vibration frequency and amplitude influence were examined and the corresponding fluid flow characteristics were investigated which add an additional complexity to the inherent problem in unsteady flow. The study was conducted for flow over NACA0012 airfoil at 600 ≤ Re ≤ 3000 at a low angle of attack. Vibration of flexible structure induced a secondary vortex which modifies the pressure distribution and lift performance of the airfoil. It was noted that, in the case of Re ≤ 1000, the deformation of flexible structure was occurred in-phase with the vortex shedding. In other words, increasing maximum lift was linked with the positive deformation of flexible structure. At Re=1500 a phase shift of about 1/π was existed while they were out-of-phase at Re>1500. In addition, the oscillation amplitude of lift coefficient increased with increasing vibration amplitude for Re≤1500 while it decreased with increasing vibration amplitude for Re>1500. As a result of frequency lock-in, the average lift coefficient was increased with increasing vibration amplitude for all investigated Reynolds numbers (Re). The pressure-velocity coupling is obtained using pressure implicit and splitting of operators (PISO) algorithm

In above equations \( A_v \) and \( f_v \) denote the vibration amplitude and frequency, respectively.

In this work, the authors did not take many ranges for the flow velocity to simulate the reality of blade velocity in the air. The study should consider effect of vibrations on the pressure side down the blade. The study did not take into account the effect of vibration by stabilizing the vibration amplitude in order to give a more clear study in the case of changing the vibration with what happens in the locked in. However, their study was in terms of representing the effect of vibrations on the distribution of pressures above the blade and thus changing the amount of CP or CL of the blade. In addition, it presented the visualization of the locked in occurring in the blade as a result of vibration and its frequency.

1.4. Experimental work

Ongoren and Rockwell [29] studied the vibration effect of cylinders of various cross-section subjected to controlled oscillations in a direction transverse to the incident flow. Excitation was at frequency \( f_e \) relative to the formation frequency of \( f_0 \) large-scale vortices from the corresponding stationary cylinder, and at Reynolds numbers in the range 584 < Re < 1300. Modifications of the near wake were characterized by visualization of the instantaneous flow structure in conjunction with body displacement-flow velocity correlations. At \( f_e / f_0 = 0.5 \), which corresponds to subharmonic excitation, as well as at \( f_e / f_0 = 1 \), the near wake structure was phase-locked (synchronized) to the cylinder motion. However, the synchronization mechanism was distinctly different in these two regimes. Near or at \( f_e / f_0 = 1 \), the phase of the shed vortex with respect to the cylinder displacement switches by
approximately π. Over a wide range of \( f_e / f_0^* \), the perturbed near wake rapidly recovered to a large-scale anti-symmetrical mode similar in form to the well-known Karman vortex street. The frequency \( f_0 \) of the recovered vortex street downstream of the body showed substantial departure from the shedding frequency \( f_0^* \), from the corresponding stationary body. It locks-on to resonant modes corresponding to \( f_e / f_0^* = 1/n \). This wake response involved strictly hydrodynamic phenomena. It showed, however, a resonant behavior was convectively unstable and was similar to that of coupled flow-acoustic systems. The frequency was introduced as a ratio \( f_e / f_0^* \). He used a hydrogen bubble to visualize the flow and vortices, it was necessary to ensure that its metallic surface was well insulated from the electrolysis process during hydrogen bubble visualization.

Von Kármán vortex karman. Although this study took more than one geometric form in a practical way and gave the description of the eddies and their decay, it did not give the pressure distribution behavior on the body \( CP \) and \( CL \), and the effect of these important factors on the bodies through which the fluid passes. Frequency ranges were not defined explicitly as they defined as quasi-vibration, no studies were presented about frequency ranges and amplitude on the vortices form. The research gave a clear presentation about the composition, number, and locations of vortices behind the body in the fluid stream.

Lifson, Smailey and Knauf, [30] presented a basis for selecting and justifying vibration monitoring equipment for power-generating gas turbines. Users of industrial gas turbines from utility and petrochemical companies were surveyed; a utility forced outage database was analyzed; typical vibration limits were presented; and the current capabilities of commercial monitoring systems and vibration transducers were summarized. The industry survey by site visits and questionnaire develops common trends; it itemizes malfunctions that can be successfully identified with appropriate vibration monitoring; it summarized current practices, benefits, limitations, and operating experience with various transducer types, as applied to harsh gas turbine environments. Vibration limits, trending, and sources of vibration were addressed. Operational factors were considered in planning and cost justifying vibration monitoring systems for a basic trip protection, periodic measurements, and on-line computerized continuous protection. Seventeen case histories and examples illustrated and supported these findings. Analysis of the utility-generated database complements the industry survey; it isolates the contribution of different vibration-related outages for base loaded and peaking units; graphic results break down these outages into duration, man-hours to repair, and frequency of occurrence.

Mathioudakis, Loukis and Papailiou, [31] presented the results from an experimental investigation of the compressor casing vibration of an industrial gas turbine. It was demonstrated that the statistical properties of acceleration signals can be linked with engine operating conditions. The power content of such signals was dominated by contributions originating from the stages of the compressor, while the contribution of the shaft excitation was secondary. Using nonparametric identification methods, accelerometer outputs were correlated to unsteady pressure measurements taken by fast response transducers at the inner surface of the compressor casing. The transfer functions allow the reconstruction of unsteady pressure signal features from the accelerometer readings. A possibility was thus provided for "seeing" the unsteady pressure field of the rotor blades without actually penetrating through the casing, but by simply observing its external surface vibrations.

Six accelerometers were mounted on the outer surface of the compressor casing. They were piezoelectric, manufactured by METRAVIB-RDS (model AM-109-MP) with a frequency range 1.5 Hz to 10 kHz, a linearity ±2 percent up to 1000g, and a minimum resonant frequency of 30 kHz. Fast response pressure transducers were flush mounted at the compressor inner casing wall. Generally, The researchers did not focus on the aerodynamics and the mechanism of pressure distribution between the compressor blades, but rather they were satisfied with the final pressure for each stage. There was no accurate descriptions of the vibration caused by the flow, nor was there an equation explaining the
effect of vibration on velocity and vice versa. The only expressed the noise caused by the vibration as
an indicator of the action. They did not give an equation or diagram of the relationship of vibration
with pressure as it being important in this work. However, the vibration ranges are widely accepted.
They presented a good idea of the nature of the pressure and the speed of the compressor at each stage
of the air pressure.

Williamson and Govardhan, [32], This review summarize fundamental results and discoveries
concerning vortex-induced vibration (VIV), that have been made over the last two decades, many of
which were related to the push to explore very low mass and damping, and to new computational and
experimental techniques that were hitherto not available. We bring together new concepts and
phenomena generic to VIV systems, and pay special attention to the vortex dynamics and energy
transfer that gives rise to modes of vibration, the importance of mass and damping, the concept of a
critical mass, the relationship between force and vorticity, and the concept of “effective elasticity,”
among other points. We present new vortex wake modes, generally in the framework of a map of
vortex modes compiled from forced vibration studies, some of which cause free vibration. Some
discussion focuses on topics of the current debate, such as the decomposition of force, the relevance
of the paradigm flow of an elastically mounted cylinder to more complex systems, and the relationship
between forced and free vibration. The authors did not consider effect of the cylinder vibration on the
flow and how the vortices formed behind the body, beside no clear calculation for the amount of CP of
the body due to its vibration. However, the good side of this work is studying the damping effect on
the vibration caused by the mass motion. A clear picture about vortices was presented. Good
information about vortices number, locations, and development was discussed in detail.

Govardhan and Williamson, [33], studied the transverse vortex-induced vibrations of a cylinder with
no structural restoring force (k = 0). In terms of the conventionally used normalized flow velocity, U∗,
the present experiments correspond to an infinite value (where U∗ = U/fsD, fs = natural frequency, D
=diameter). A reduction of mass ratios m∗ (mass/displaced mass) from the classically studied values
of order m∗ = 100, down to m∗ = 1, yields negligible oscillations. However, a further reduction in mass
exhibits a surprising result: large-amplitude vigorous vibrations suddenly appeared for values of mass
less than a critical mass ratio, m∗ crit = 0.54. The classical assumption, since the work of den Hartog
(1934), has been that resonant large-amplitude oscillations exist only over a narrow range of
velocities, around U∗ ~ 5, where the vortex shedding frequency was comparable with the natural
frequency. However, in the present study, authors demonstrated that, so long as the body’s mass was
below the critical value, the regime of normalized velocities (U∗) for resonant oscillations was
infinitely wide, beginning at around U∗ ~ 5 and extending to U∗ → ∞. The result was in precise
accordance with the predictions put forward by Govardhan and Williamson (2000), based on
elastically mounted vibration studies (where k > 0). The author deduced a condition under which the
unusual concept of an infinitely wide regime of resonance occurred in any generic vortex-induced
vibration system.

He used a Digital particle image velocimetry (DPIV) to visualize clear behavior of the flow
including vortices. Although the author presented a hard and good work in practical and theoretical
terms, he did not address a very important topic, which is the effect of pressure distribution
with/without vibration, as well as pressure and lift coefficient cp and cl. No clear picture was given
about the effect of vibration. However, the results were accurate and most of the flow characteristics
and its effect were about the vibration effect. The distribution of flow and velocity was clearly visible
to observe the regions of separation and formation areas on the body.

Gostelow, Carscallen and Platzer, [34] demonstrated similarities between the vortex shedding from
blunt trailing-edged transonic turbine nozzle blades and from oscillating bluff bodies. Under subsonic
conditions the turbine nozzle cascade shed wake vortices in a conventional von Kármán vortex-street.
This was linked with a depressed base pressure and associated energy separation in the wake. Under transonic conditions a variety of different shedding configurations was observed with vortices shedding and pairing in several different ways. Similarities were addressed between the observed structures and those from vortex shedding in some other physical situations. The authors investigated the similarity between the vortex wakes shed from cylinders and airfoils in sinusoidal heaving motion in low-speed flow and the wakes shed from the turbine nozzle cascade in transonic flow. The established field of vortex-induced vibration provided a developed classification scheme for the phenomena observed. The paper brought together three previously unrelated fields of investigation and, by showing that the three were essentially related, provided the basis was for a new synthesis. It is important to mention that the author showed a good regularity of the vortices representation and their locations, and showed a good map to the number of eddies, their streamlines, and their distribution behind the vibrating body in the fluid flow, the regularity of the representation of vortices, and areas of decay of eddies clearly. In addition, the research was limited to the practical part and to make an analogy between the formation of vortices and their decay between blade and cylinder. Furthermore, the author did not include numerical solutions or simulations of the real case.

Presented the results of an analysis of the gas-dynamic control of flow around the guide blades of axial compressor stage for the vibration intensity of rotor blades. It was shown that gas-dynamic action on the flow around blades equalized the flow velocity field before the rotor and reduced the level of resonant stresses in blades in the case of their resonant excitation. The total aerodynamic force that acts on the blade in the plane of vibrations is given by

\[ P_b = P_{a_g} \cos \theta + P_{a_b} \sin \theta \]  

(13)

In this work, the extent of the effect of vibration on the aerodynamics was not shown through the results or the body contours for the distribution of velocity and pressure. The engineering analysis of practical modeling was good. In addition, the use of the simulation program and the method of their analysis from the equations adopted in this program were fairly good.

Hsieh, Low, and Chiew, [35] investigated the flow characteristics of the wake in the initial branch for a vibrating cylinder using a new PIV technique capable of a high sampling rate of 200 Hz. The experiments were carried out over a wide range of reduced velocities to better understand the important trends. The mean velocities and turbulence characteristics were obtained by ensemble averaging repeated velocity measurements. Their study revealed new insights on the flow characteristics of the wake in the initial branch. In particular, the cylinder vibration was found to lead to the formation of oblique jets, which have a profound influence on the mean flow velocity, turbulence intensity, formation and convection of the vortices, and variations to the stagnation and separation points of the cylinder during vibration. The PIV results of a vibrating cylinder with \( u_0 = 18.76 \text{ cm/s (Vr = 6.66)} \) under different sampling rates (10, 33.3, 100 and 200 Hz).

He used a laser projector to show the shadow regions. The authors did not show clear and explicit interest in the aerodynamics affected by the vibration, although they took the formation of vortices behind the cylindrical body and was able to represent these vortices well. The author focused on making a map of those vortices, their number and direction.

Dahham and Alkhafaji, [36] focused on studying the behavior of velocity profile under force vibration for a different frequency (34, 48, 65 and 80 Hz) for lower annulus of Can Combustor. An experimental rig was designed at Babylon University /Iraq by the author. The Can Combustor tested in this study was real part collected from Al-Khairat/Iraq gas turbine power station. The velocity profiles were examined at three positions in the annular for lower region. The velocity in X-direction calculating with a different frequency for lower annulus. The results were shown that the increase of frequency leads to an increase the velocity profile and large recirculation zone will form in some
points. Since the flow was turbulent, so the slope of the velocity profile at the wall was big but when forced vibration effect was applied it becomes much greater than without vibration effect about (25%). Reynolds number increasing with praise of velocity in X-direction. Also, the increase in vibration level produced a non-uniform velocity profile which affected the spreading of cooling efficiency. Finally, the shape of the velocity profile changed from flatter to non-uniform shape due to fluctuation vibration effect so, the cooling film air fails to protect the linear wall of Combustor from damage.

Dahham and Alkhafaji, [37] concentrated on studying the behavior of velocity profile under the influence of different frequency (34, 48, 65 and 80 Hz) in each of the upper and lower annulus of Can Combustor. An experimental rig was designed to simulate the annulus flow inside a Can Combustor. The Can Combustor tested was real part collected from Al-Khairat/Iraq gas turbine power station. The velocity profiles were investigated at three positions in the annular for upper and lower region. The axial velocity and turbulence intensity were calculated with different frequency for upper and lower annulus. The results were shown that the increase of frequency lead to increase the velocity profile and large recirculation zone will build in some points. Reynolds number increased with raise of axial velocity. Also the increasing in vibration level caused a non-uniform velocity profile which affected the distribution of cooling effectiveness. Different frequency (34, 48, 65 and 80 Hz) were studied in this work.

1.5. Theoretical and Experimental Work

Jungst et al. [38] added to the understanding of non-synchronous blade vibration caused by unsteady flow close to the stability limit of transonic compressor rotors. Blade vibrations were measured in the rotating frame of reference by strain gauges applied to the blades and additionally in the stationary frame of reference by capacitive tip clearance/tip timing sensors. Furthermore unsteady pressure transducers in the casing wall were used to analyze the flow phenomena during high blade vibration amplitudes. During the transient experiments, the compressor was back-pressured into its stability limit. Two transient measurements of a compressor setup with an enlarged tip clearance of 2.5% blade chord were analyzed at part speed and design speed. This means that subsonic and transonic operation conditions were analyzed. In both cases, nonsynchronous vibration occurred and limited the compressor operation. The mechanism leading to these vibrations was based on an unsteady flow pattern that rotates relative to the rotor, reported as rotating instability. The author did not show the effect of the vibrations on fluid flow and aerodynamics in terms of the distribution of pressure $C_P$ and lift $C_L$ coefficients, as these parameters are important in giving an insight into how the performance blades in the compressor. As a result of the vibrations, the author focused on the flexibility of the blades under vibration only. However, the simulation study of the unsteady flow was difficult to implement because of their changes in the fluid properties over time. The authors presented a clear vision of the pressure distribution around the compressor blade in a good theoretical procedure. In addition, the author presented a clear map of pressure in three regions of the blades.

Moller et al. [39] presented a numerical study on blade vibration for the transonic compressor rig at the Technische Universitat Darmstadt (TUD), Darmstadt, Germany. The vibration was experimentally observed for the second eigenmode of the rotor blades at nonsynchronous frequencies and was simulated for two rotational speeds using a time-linearized approach. The numerical simulation results were in close agreement with the experiment in both cases. The vibration phenomenon shows similarities to flutter. Numerical simulations and comparison with the experimental observations showed that vibrations occur near the compressor stability limit due to interaction of the blade movement with a pressure fluctuation pattern originating from the tip clearance flow. The tip clearance flow pattern travels in the backward direction, seen from the rotating frame of reference, and causes a forward traveling structural vibration pattern with the same phase difference between blades. When decreasing the rotor tip gap size, the mechanism causing the vibration was eased. The SGs were sampled with a frequency of 100 kHz, the BTC/BTT sensors have a sampling rate of 2500 kHz, and
the WPTs record data with a frequency of 500 kHz. The signals of the WPTs are low-pass filtered at 100 kHz.

Dahham, Alkhafaji and Al-jelawy, [40] This phenomenon has been studied experimentally and numerically. The Computational Fluid Dynamics analysis was accomplished by utilizing the Shear-Stress Transport (SST) k-omega model to predict the flow velocity at the recirculation zone. The vibration testing equipment was designed and used to apply the excitation forces on the wall combustor. It has been explained that the reversed flow which causes eddies inside the recirculation region can be increased at higher frequencies. In addition to that, exciting the system with higher frequencies would increase the turbulence intensity causing a recirculation region enlargement. The Computational results were compared against the experimental results, and they show a very good agreement. On the other hand, the static pressure distribution has been decreased while increasing the frequency. It has been proved that the frequency values play an essential role to predict the system behavior. It is important to mention that the author did not show a clear visualization of the Mach number distribution in order to know if there was a shock that might occur in the study area or not. Nest, no clear explanation of the characterization of high vibration ranges was performed in the simulation, as it is a constant value or a variable equation with the fluid velocity. The study showed clear visualization of the velocity and pressure distribution of the amplitude vibration ranges. There was congruence in the equations that govern the formation of vortices in turbulent flow with the k-w function.

2. Conclusions
The highlight in this paper is a survey of aerodynamic research in the field of the effect of vibration on the aerodynamics of the flow and what happens to the flow structure of the changes, including useful and others that are not useful according to both the extent and amplitude of vibration. Most of the studies that have been presented have given somewhat acceptable results to form a perception of the dynamic behavior of the fluid flow around bodies in general and in the gas turbine system in particular. Where some research has dealt with the distribution of pressure, speed, and Mach number in the form of a contour, and this is one of the important things in highlighting the aerodynamic performance of the fluid around the body. Some research has also taken the aspect of showing the formation of eddies and concern for their number, locations, and how those eddies decay. However, this research went beyond showing flow characteristics such as elevation, pressure, and Mach number. Some of these studies dealt with the presence of vibration occurring on the body and the calculation of the extent to which the aforementioned aerodynamic properties are affected by these vibrations. Also, it was noticed through the diagrams and figures, the extent to which they were affected by the amount of vibration and the amplitude of the vibration, as the researchers took different ranges.

An overview of the development of fluid flow models around the bodies and a survey of recent research on aerodynamic investigations of turbine blades are presented. The effects of fluid flow phenomena are described as they affect the aerodynamic performance of the series turbine stream flow. Development of flow between the blades and passage flow. Descriptions and models of the proposed fluid flow were discussed by a variety of investigators, including implications for local and overall aerodynamic determination. The formation, trend and number of vortices are also considered. In general, results from several studies show that boundary layers, and wake all contribute to overall optimum performance in relative values depending on the Mach number presented as a body contour for passage, vortices, and angular vortices that arise within the flows that develop along the inner walls. The most important parameters affecting the initiation and development of such flows within the turbine stages are the structure and geometry of the interior walls.
3. Nomenclature

- $V_d$: Velocity of propagation of disturbance (always positive)
- $f(t)_r$: Fourier series expansion of this force on the blade
- $0$: Value at the origin
- $1$: Upstream of cascade generally
- $2$: Downstream of cascade generally.
- $A$: Constant.
- $H$: is the Heaviside function and
- $P$: Total pressure.
- $U$: Mean velocities of flow
- $c$: Blade chord
- $i$: Indicates component leading $90^\circ$ in phase.
- $s$: Cascade spacing.
- $x(t)$: motion of the blade.
- $x, y$: Rectangular co-ordinates
- $\alpha$: Angle of air flow
- $\zeta$: vorticity
- $\theta$: is the angle between the direction of vibration and a normal to the front of cascades
- $\lambda$: Frequency parameter.
- $\rho$: fluid density
- $\psi$: stream function
- $\omega$: Angular frequency of vibration

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