EXPERIMENTAL AND NUMERICAL INVESTIGATION OF THE CAVITATING DRAFT TUBE VORTEX IN A FRANCIS MODEL TURBINE

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ABSTRACT

The accumulated experience, their increased reliability as well as the possibility to cover a large range of specific speeds have contributed in the wide use of Francis turbines for energy generation. Recently, turbines are allowed to operate in a wider range of flow conditions, in response to the increasing flexibility needs of electric grids. At part load and full load conditions, however, there is a possibility of cavitating vortex core formation at the outlet of the runner, which is associated with large pressure pulsations and flow instabilities that limit the operating envelope of the turbine. In order to study the effects of draft tube vortex cavitation, experimental and numerical work took place. The inception and development of draft tube vortex cavitation was identified and tracked at various operating conditions, and the results were compared to numerical predictions. For the numerical analysis of the turbine the unsteady RANS equations were solved using commercial CFD software. Dynamic pressure measurements as well as vibration measurements were obtained at the draft tube cone using dynamic piezoelectric transducer and piezoelectric accelerometers, respectively. The pressure and vibration results were analyzed both in time and frequency domain and good correlation between the experimental and numerical values, was achieved. The numerical analysis of draft tube cavitation can be utilized as a useful tool in the early stages of a Francis turbine design procedure, in order to extend its operating range. In addition, the present approach and obtained results can be used for the determination of an acceptable operating regime within regions where the vortex core appears, coupled to condition monitoring of existing Francis turbines.

KEYWORDS

Francis turbines, draft tube vortex core, vibration

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1. INTRODUCTION

Significant work has been done in predicting the helical vortex in the draft tube of Francis turbines. The helical vortex can cause low frequency pressure pulsations which can lead to severe fluctuation and damage in the structural integrity of the turbine. It is therefore important to be able to closely monitor under varying operating conditions physical quantities of interest that are related to this phenomenon. Turbine operators, in order to maximize energy productivity tend to operate turbines at off design conditions and at lower flows. This fact results in working the turbine in regions where strong instabilities occur. One important instability detected in Francis turbines is the formation of the vortex rope cavitation, which extends from the hub of runner until the end of draft tube. The vortex rope is related with the existence of the tangential component of the absolute velocity at the exit of runner, when the machine operates either in part or over load conditions. As a consequence vortex instabilities could not only cause flow induced power swings, but also its frequency could excite the natural frequency of either the turbine or the powerhouse. Recent years, many researchers have focused in the experimental and computational study of this phenomenon in order to understand its physical behavior and to achieve its treatment.

In one of the earliest studies pressure fluctuations were measured in order to depict the variation of pressure with power [1]. It was observed that the intensity of pressure decreases only for 0.82~0.92 of maximum power. Pressure data in specific operating conditions underwent further processing in order to identify the instability that drives the frequency. Finally, air admission provided under the runner and significant reduction in pressure intensity has been observed in a wide range of power generated. In addition to previous measurements, Bajic [2] measured the noise that was emitted from each guide vane in a Francis turbine and he used an efficient way to process the signals obtained. This study proposed the subtraction of the environmental noise from the power spectrum and the decomposition of the remaining spectrum in three different load mechanisms. By this way, it is possible to diagnose successfully different types of cavitation in a turbine. For the purpose of studying different types of cavitation in Francis turbines and for finding the most adequate placement of the sensors, not only pressure fluctuations but also vibrations were measured in [3]. Pressure transducers were placed in two places; on the draft tube and at the upstream of the runner, and vibration was measured by two accelerometers that were mounted in turbine’s guide bearing. According to the results, it was not possible to diagnose vortex rope by taking measurements on the guide bearing as well as more accurate pressure results were acquired at draft tube than upstream the runner. Meanwhile research was focused on pressure variation due to vortex rope in draft tube when the turbine is operated at high part load conditions [4]. Pressure data were obtained in the direction of recognizing the mechanism that caused high pressure values and in the end it became possible to relate strong pressure fluctuations with the elliptical shape of the rope.

At the same time, in [5] the response of one turbine that was installed between two prototype Francis turbines was investigated, by measuring vibrations with pressure transducers and strain gauges. Furthermore, average and standard variation values of pressure and von Misses stresses were calculated, measured in different operating conditions and in the end, the operating point with the strongest vibrations was identified. In accordance to the results of Zhukovskii [1] it was proved that air injection managed to decrease pressure and stresses fluctuations. Analysis of vortex rope was focused at higher part load conditions implementing pressure sensors and a high speed camera in [6]. The results proved the elliptical shape of vortex rope as well as the strong relation of pressure fluctuation with cavitation number.

Meanwhile, mechanical vibration sources occurred to the shaft were added in the experimental analysis in [7]. Proximity sensors were used for measuring mechanical vibrations and accelerometers for measuring hydraulic vibrations. After the analysis of the results it was verified that flow instability was reduced along the length of draft tube.

Apart from experimental work, numerous computational investigations have been presented, studying the flow behavior in the draft tube and the appearance of the vortex core in part load and full load conditions. The effect of turbulence modeling, domain configuration and mesh density have been presented in [9]. By comparison to experimental data it was shown that the frequency spectrum can be adequately captured even by using moderate grids, while at the same time, a significant discrepancy in the amplitudes was observed. Similarly, in [10], it was shown that an acceptable agreement between numerical and experimental results can be achieved. In order to fully capture the physical phenomena taking place in a flow field with the processing vortex rope, cavitation modelling has often been used [11,12]. However, it has been shown that including a cavitation model in computations has little effect on the prediction of the frequency domain, as opposed to again the predicted amplitudes [9, 11], which are dependent on the cavitation number.

According to all the above work discussed the significance of pressure and vibration measurements in order to study vortex rope instability in Francis draft tube is evident. Also it is necessary to develop computational tools that are able to predict the physical characteristics of vortex rope. As a consequence this paper deals with the experimental and numerical study of vortex rope on the draft tube of a Francis turbine model that is installed in the laboratory of Hydraulic Machines at National Technical University of Athens. Experimental measurements were obtained by a piezoelectric pressure transducer and piezoelectric accelerometers and the numerical simulations were performed by commercial Computational Fluid Dynamics (CFD) software.

2. TURBINE CHARACTERISTICS AND EXPERIMENTAL SET-UP

The model of Francis that is used in this study is presented in fig. 1. It is a low specific speed turbine that is capable to produce 7 kW power at 9m head with 0.08 m^3/sec flow rate at its best efficiency point. A centrifugal pump that is also installed at the laboratory with specific speed n_q=1410, provides the available head by pumping water from laboratory's water tank.

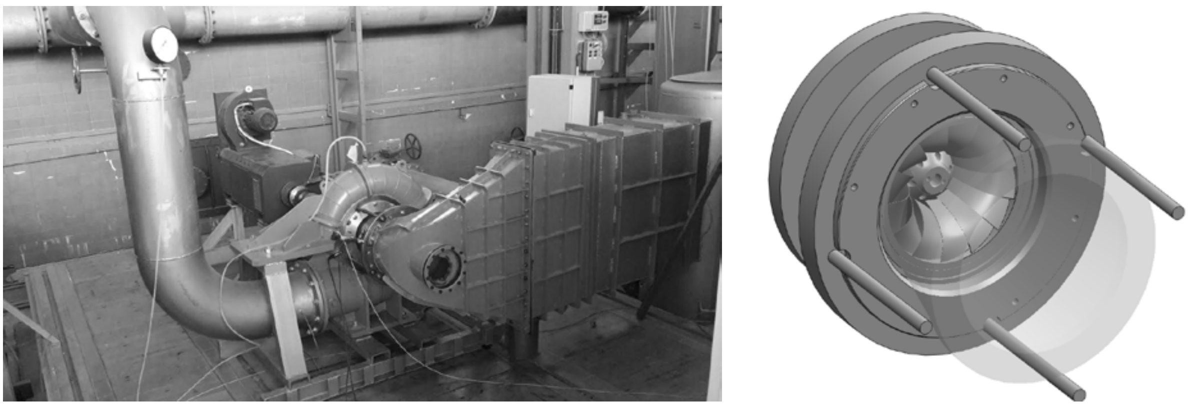

Figure 1 Francis turbine

The flow passes first from the spiral case with the 12 stay vanes, next from 24 guide vanes and finally enters the runner that is made up from 11 vanes, where the energy is produced.

The angle of guide vanes is changed by using an external mechanism in order to change the volume flow rate. In this experiment it is critical to change the operating point of the machine in order to study acceleration of vibration in partload conditions. The operating points that were chosen are presented in Table 1:

A/A	900 rpm
1	0.67 * Q_{opt}
2	0.87 * Q_{opt}
3	1 * Q_{opt}

Table 1 Operating points

The rotational speed that has been chosen is 900 rpm. According to Dörfler [13] the frequency of vortex in part loads is 0.2~0.4 of the rotational frequency, so the range of frequencies that is expected lays between 3 and 6 Hz. As a consequence two piezoelectric accelerometers and one pressure transducer which are able to measure the low frequencies that vortex rope excites have been selected**.** The accelerometers have measurement and frequency ranges equal to [±50g] and [0.58~5000Hz], respectively. Also the pressure transducer has a measurement range equal to [0~10bar] and natural frequency 12 kHz. All sensors were mounted in 180° on draft tube as presented in fig. 2. The sampling rate is selected equal to 150 Hz which is equal to ten times the rotational speed of the runner. Finally, sampling time is set equal to 6.5 sec in order to efficiently record the low frequency phenomenon.

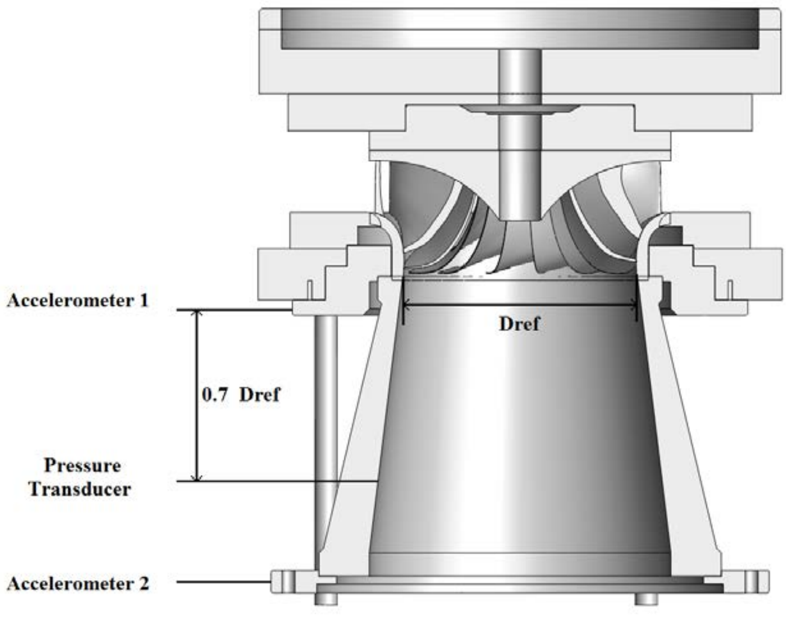

Figure 2 Sensors Positioning

3. COMPUTATIONAL MODELLING

For the numerical analysis of the precessing vortex rope, the computational model shown in fig.3 was used. In order to minimize the computational resources needed for this study, the computational domain consisted of the runner and the suction cone of the draft tube, while all other turbine components were ignored. In previous work, it has been shown that such simplifications will not have an effect on the prediction of the unsteady characteristics of the flow [14]. Previous work, has stressed the dependency of the flow evolution in the draft tube on the inlet boundary conditions [15]. In order to ensure that the correct boundary conditions are imposed at the inlet of the suction cone, the calculations included the runner. The numerical grid, shown in fig. 3, consisted of 8 million tetrahedral elements with refinement close to the walls. The boundary conditions used corresponded to the guide vane opening used in the experiments and for the outlet boundary condition, a zero static pressure was imposed. The solution was initialized using a steady state calculation using the MRF model. Upon convergence, the simulation was switched to an unsteady calculation and continued until a time periodic steady state was achieved.

For the turbulence modeling the realizable k-e turbulence model was used while for the velocity and pressure coupling, a fully coupled method was used. For all equations a second order discretization scheme both in space and time were considered. Finally, a time step corresponding to a 6^o rotation per time step was considered. Calculations were performed for various operating conditions ranging from 6mm to 12 mm guide vane opening and flow rates from 180 m^3/h to 280 m^3/h. By setting the flow rate at the runner inlet and the rotational speed of the runner a constant net head parameter could be achieved. Table 2 summarizes the operating conditions considered.

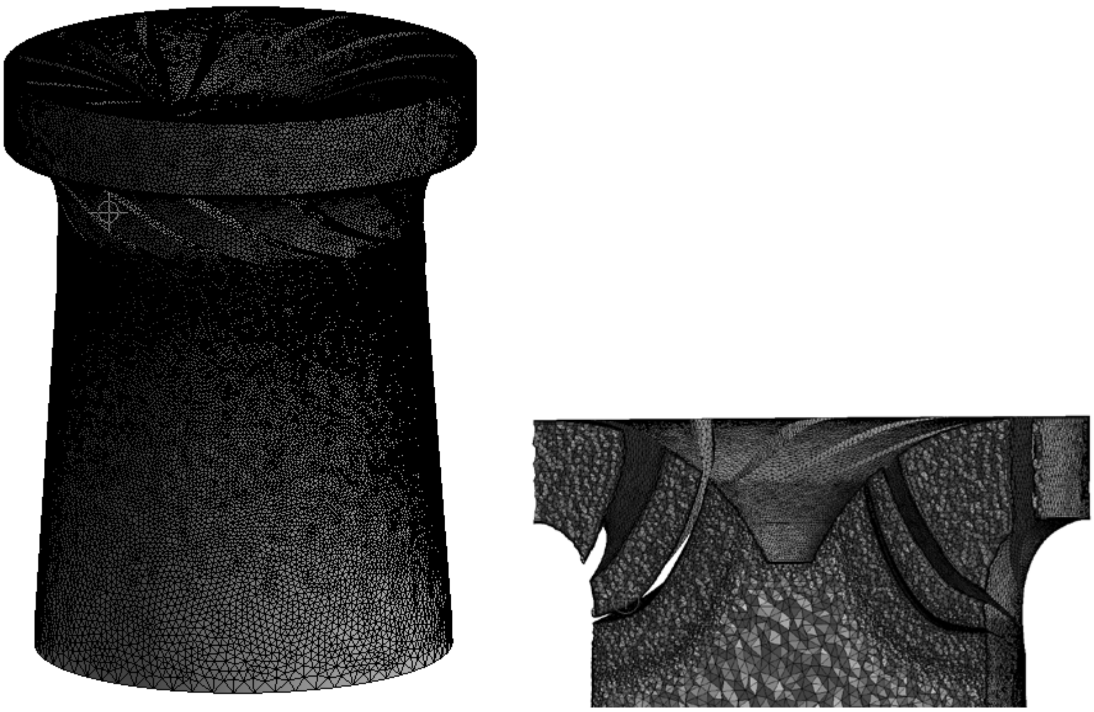

Figure 3 Computational mesh of current study(left: full domain, right: runner section detail view

4. RESULTS

At this section, the experimental and computational results are presented and discussed. For all the results presented, the horizontal axis refers to the non-dimensional relative frequency, f. Firstly, in fig. 4 the frequency spectrum of acceleration of vibration of accelerometer 1 is shown in three different flow rates. At the optimum flow rate (fig. 1a), the flow exits the runner without the tangential component, so there is no frequency component for f<1. However, the rotational frequency and mainly its second harmonic appear in the spectrum. The main reason is that the accelerometer 1 is mounted at the casing of the turbine as it is shown in fig.2, so it is able to detect mechanical vibrations. The increase of amplitude of second harmonic compared to the synchronous speed is related to the detection of parallel misalignment. As the flow rate decreases, the frequency of vortex rope appears at 0.28 times the rotational speed and its frequency amplitude increases. In addition, when the flow rate reaches 68% of optimum flow rate (Q_{opt}) the rotational speed and its harmonics cannot be observed in the frequency spectrum.

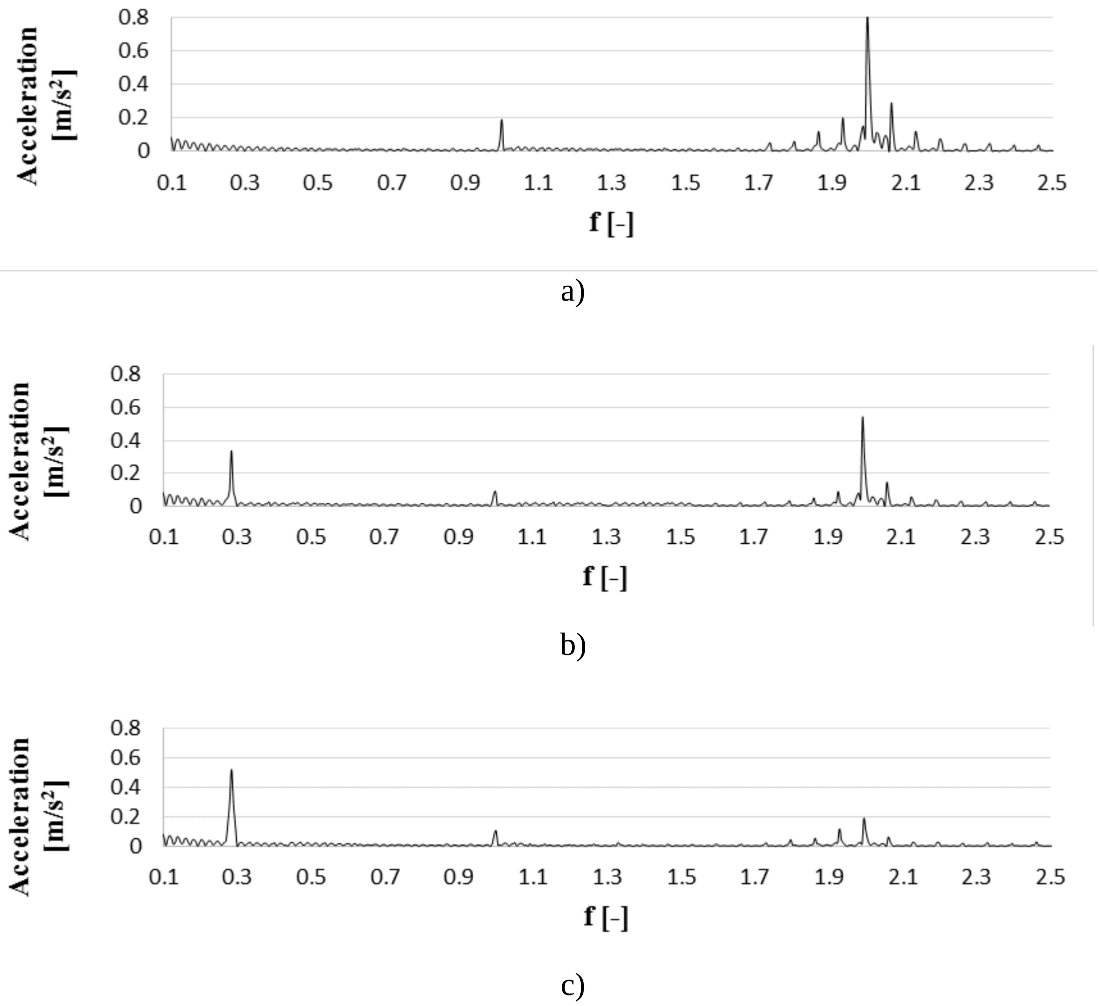

Figure 4 Frequency domain of vibration measurements in accelerometer 1, a) for $Q=Q_{opt}$, b) for $Q=0.87Q_{opt}$, c) for $Q=0.67Q_{opt}$

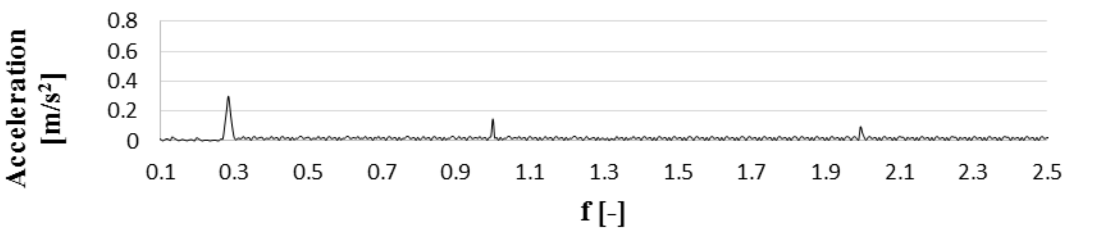

Figure 5 Frequency domain of vibration measurements in accelerometer 2 for $Q=0.67Q_{opt}$

Secondly, the frequency domain of accelerometer 2 is given at the same operating condition with accelerometer 1 in fig. 5. As in fig. 4c, the frequency of vortex rope appeared but with a slight decrease in its amplitude. The main reason of this decrease is the different position of accelerometer 2, which stands at the end of the conical draft tube. It is concluded that vibration amplitude has its highest value in the exit of runner and decreases along the draft tube. In addition, the frequency spectrum for the pressure measurements is shown in fig. 6, for the same operating conditions. On this graph, the vortex rope frequency appears at the same frequency with vibration measurements but the component of synchronous frequency and its harmonics are not shown. The good agreement of pressure and acceleration results is significant, because it confirms not only the existence of the vortex rope but also the frequency that it excites.

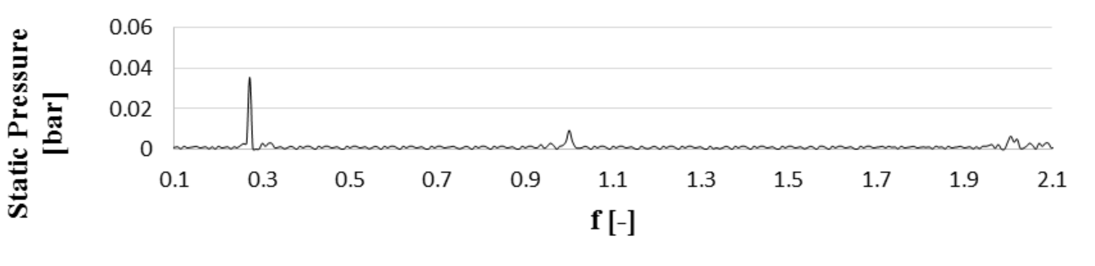

Figure 6 Frequency domain of pressure measurements for Q=0.67Q_{opt}

In fig. 7 the frequency spectrum of the pressure data from the computational modelling is presented for the same operating point. The frequency of the vortex rope is shown up at 25% of rotational speed but its amplitude is quite higher than in experimental results. The reason for this amplitude disagreement is the non-ideal conditions of the experimental measurements, although the agreement between numerical and experimental results is acceptable.

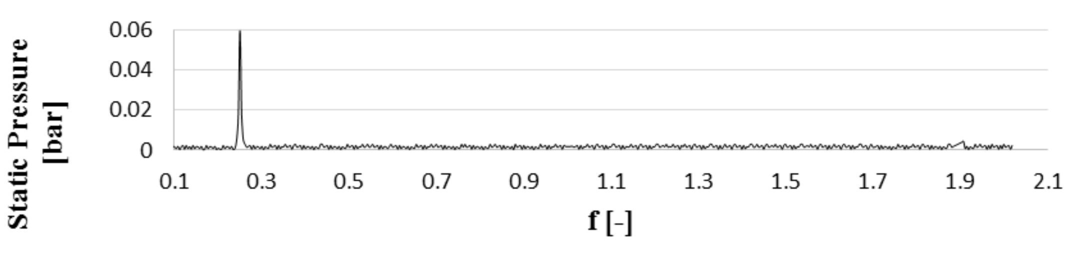

Figure 7 Frequency spectrum of the pressure data obtained from computational analysis for Q=0.67Q_{opt}

5. CONCLUSIONS

Vibration measurements in combination with pressure measurements and CFD are able to give reliable results on the field of cavitating draft tube vortex diagnosis. The vortex rope frequency has been detected at 25% and 28% of runners's rotational speed in computational and experimental results, respectively. In addition, the positioning of the accelerometers in the entrance and exit of draft tube showed that the intensity of the vortex rope decreases along draft tube. Further study of the experimental results could be utilizied in the development of condition monitoring systems appropriate to diagnose flow instabilities in hydroelectric plants. Finally, the numerical model proved reliable to model vortex rope as there were no significant differences between the numerical and experimental results. This indicates that similar relatively simple analyses can be used in the early design stages of new turbines aiming at the extension of the allowable operating range of the turbine.

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