ICSV14
Cairns • Australia
9-12 July, 2007

14th International Congress on Sound & Vibration

EXPERIMENTAL EVALUATION OF THE MODAL DAMPING OF AUTOMOTIVE COMPONENTS WITH DIFFERENT BOUNDARY CONDITIONS

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Abstract

In this paper an Experimental Modal Analysis (EMA) program has been carried out on different types of brackets used on diesel engines, and in different boundary conditions with the aim of collecting the damping values in a database. In particular, the EMA has been carried out on brackets in the free-free condition and in the actual clamped condition, highlighting the damping variation with the different boundary conditions. The results of the experimental measurements have been processed with the modal analysis solvers LSCE (Least Squares Complex Exponential) and PolyMAX using the software LMS Test.Lab.

1. INTRODUCTION

The identification of damping is essential for the computation of stress and strains in vibrating structures; in fact, the actual vibration amplitude is strongly influenced by the damping of the structure under testing. This parameter is often obtained from FRF measurements processed using curve fitting techniques or modal analysis algorithms. Unlikely, in the virtual prototyping phase, the first hard prototype is not obviously available. It is therefore necessary to look for suitable damping values in literature, which are not generally available for complex mechanical components. This is the case considered in this paper, concerning brackets used on diesel engines. Moreover, modal damping, as well as the natural frequencies and mode shapes, depends not only on the materials and the geometrical shape, but also on how the bracket is clamped to the surrounding structure (usually the engine block). Therefore, it is useful to collect a database containing modal damping values for different types of brackets, with different kinds of boundary conditions, in order to use such values in the virtual prototyping phase to analyse the dynamic behaviour of new similar brackets for which the actual prototype is not available yet. For such a reason, an experimental modal analysis (EMA) program has been carried out on different types of brackets with different boundary conditions (free-free, clamped, clamped with accessories) with the aim of collecting the damping values in a database. These brackets (that will be identified with the capital letters A, B, C and D) are used in diesel engines to support different engine components such as gear pumps for steering systems, water pumps, oil pumps, gas turbines, etc. Bracket A is used in marine diesel engines to support the water pump for the cooling system; bracket B and

bracket D are used in automotive engines to support the pump for the steering system and finally bracket C is used in automotive engines to support the alternator, fuel pump and water pump [1]. The bracket material is aluminium.
The structure of the paper is as follows: Section 2 - introduction to experimental setup, Section 3 - illustration and discussion of the results of the modal analyses in terms of natural frequencies and modal damping; Section 4 - conclusion.

2. EXPERIMENTAL SETUP

For modal testing, it is necessary to measure the Frequency Response Functions (FRFs). For this reason an impact hammer (PCB 068C04) has been used to excite the different measurement points of the brackets and PCB piezoelectric accelerometers (frequency range 1 to 10000 Hz) have been mounted on the brackets in order to measure the responses. The measurement locations were chosen in order to give an adequate spatial resolution to describe the global structural mode shapes. The procedure used to perform the EMA is the conventional procedure in which both excitation and response are measured simultaneously to obtain the *Inertance*, i.e. the FRF between acceleration and force. In particular, the response points were maintained fixed during the tests, while the excitation moved from one measurement point to another in order to obtain the FRFs among all the considered points. The signals were acquired by using sampling frequency and frequency resolution according with the type of brackets and with the kind of boundary condition; furthermore the FRFs are calculated by using the H_1 estimator available in LMS Test.Lab [2]. Table 1, Table 2, Table 3 and Table 4 collect the acquisition parameter setup for the brackets. In particular, these tables show bandwidth, frequency resolution, number of excitation points, number and types of accelerometers and the number of averages for FRF estimation.

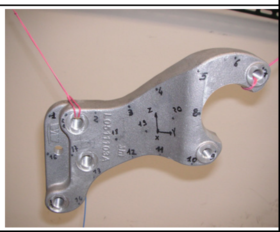
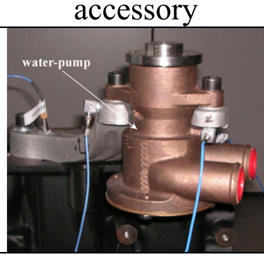
<u>Bracket A</u>	Free-free	Clamped	Clamped with accessory
			
Bandwidth	10240 Hz	10240 Hz	8192 Hz
Resolution	10 Hz	10 Hz	0,5 Hz
# excitation points	21	21	21
Accelerometers	One PCB353B21 piezoelectric tri-axial accelerometer	One PCB353B21 piezoelectric tri-axial accelerometer	Two PCB353B21 piezoelectric tri-axial accelerometers and two PCB353B18 mono-axial accelerometers
# averages	10	10	10

Table 1 : setup for bracket A

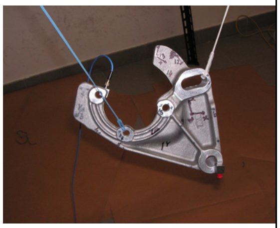
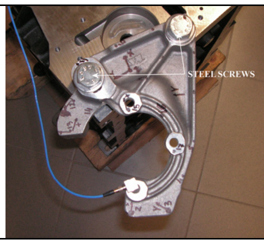
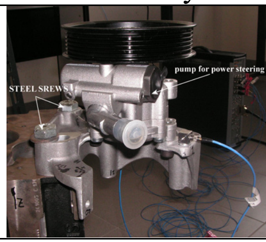
Bracket B	Free-free	Clamped	Clamped with accessory
			
Bandwidth	16384 Hz	8192 Hz	8192 Hz
Resolution	1Hz	1Hz	1Hz
# excitation points	22	22	22
Accelerometers	Two PCB353B21 piezoelectric tri-axial accelerometers	One PCB353B21 piezoelectric tri-axial accelerometer and one PCB353B18 mono-axial accelerometer	One PCB353B21 piezoelectric tri-axial accelerometer and one PCB353B18 mono-axial accelerometer
# averages	10	10	10

Table 2 : setup for bracket B

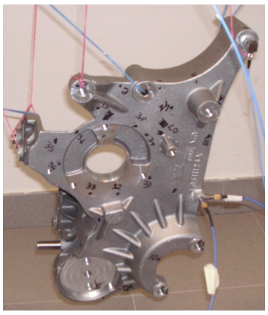
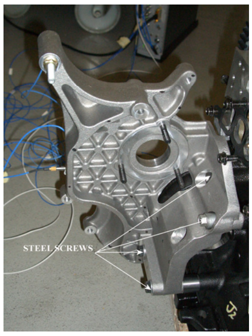
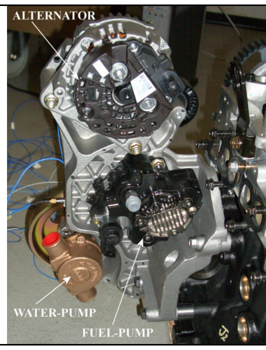
Bracket C	Free-free	Clamped	Clamped with accessories
			
Bandwidth	10240 Hz	10240 Hz	20480 Hz
Resolution	10 Hz	1,25 Hz	1,25 Hz
# excitation points	41	41	41
Accelerometers	One PCB353B21 piezoelectric tri-axial accelerometer and two PCB353B18 mono-axial accelerometers mounted in the same point	One PCB353B21 piezoelectric tri-axial accelerometer and two PCB353B18 mono-axial accelerometers mounted in the same point	One PCB353B21 piezoelectric tri-axial accelerometer and two PCB353B18 mono-axial accelerometers mounted in the same point
# averages	10	10	10

Table 3 : setup for bracket C

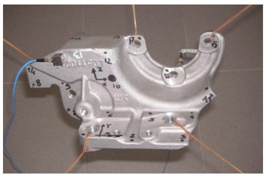
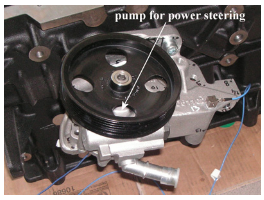
Bracket D	Free-free	Clamped	Clamped with accessory
			
Bandwidth	8192 Hz	4096 Hz	4096 Hz
Resolution	1 Hz	1 Hz	1 Hz
# excitations points	21	21	21
Accelerometers	One PCB353B21 piezoelectric tri-axial accelerometer and two PCB353B18 mono-axial accelerometers	One PCB353B21 piezoelectric tri-axial accelerometer and one PCB353B18 mono-axial accelerometer	One PCB353B21 piezoelectric tri-axial accelerometer and one PCB353B18 mono-axial accelerometer
# averages	10	10	10

Table 4 : setup for bracket D

For each measurement location, the FRFs are acquired in three orthogonal directions in order to estimate the mode shapes in the 3D space. The input auto-power-spectrum, output auto-power-spectrum and cross-power-spectra are evaluated and stored for each measurement location. An exponential window for the accelerometer signals and a force window for the force signals are used in order to reduce leakage. During the tests, the coherence function is monitored as an on-line check of the data quality. As well known, the coherence function gives a measurement of the degree of linear dependence between two signals as a function of frequency; a coherence value less than 1 highlights noise in the measurement or a non-linear behaviour between two measured signals composing the FRF.
As said, the brackets were tested on three different kinds of boundary conditions. The free-free condition was approximated by suspending the brackets using soft bungee cords. The natural resonances of the bungee cords are much lower than the lowest resonance frequencies of the brackets, so the modes of the bungee cords do not influence the analysis of the brackets. The brackets on the clamped conditions were analysed by screwing them to their engine block as in the actual condition. Finally, in the last scenario the accessories were fixed upon the brackets, clamped to the engine block. The total mass of the additional accessories of the brackets is 5 kg (water pump) for bracket A, 4.2 kg (pump for steering system) for bracket B and D, 19 kg (water pump, fuel pump, alternator, two bearings and one tension pulley) for bracket C.
The different brackets were fixed on different engine blocks, concerning 6 cylinder or 4 cylinder engines, as in the actual configuration. An EMA of these two engine blocks was also performed in order to estimate their modal properties. Such an EMA is important for the evaluation of the effective mode shapes of the brackets. In fact, some resonance frequencies of the brackets were observed as being close to natural frequencies of the engine block, moreover the corresponding mode shapes of the brackets do not show deformation but only rigid displacement in all the measurement points. Therefore, these mode shapes, which include the deformation of the engine block at which the brackets are fastened, are not their own mode shapes of brackets. Moreover it has to be underlined that we are interested in evaluating the modal damping of brackets for the analysis of new prototypes; thus, the data

which are strongly influenced by the modal properties of a specific engine block are not to be considered. For this reason, in the following section, some mode shapes will be neglected because they do not deal with the brackets, but they are due to the engine block.

3. RESULTS AND DISCUSSION

Once the experimental modal tests and analyses of the brackets have been performed, frequencies, damping and mode shapes are available for all modes in the frequency band of analysis. Table 5, Table 6, Table 7 and Table 8 show the frequency band of analysis, the stabilisation diagram obtained by using LSCE solver overlapped with the FRF sum, and the natural frequency (f_n) and modal damping (ζ) obtained by averaging the values coming from the LSCE and PolyMAX methods. Table 5 also depicts the power spectrum density (PSD) of the excitation force concerning all the measured points.
The results of the analyses show that small differences in natural frequencies and modal damping occur in the same modal test by using the two LSCE and PolyMAX solvers. Thus, the MAC (Modal Assurance Criterion) - a technique to determine the degree of correlation between the mode shapes [3] – has been applied to the corresponding eigenvectors, estimated by LSCE and PolyMAX solver: the MAC values are close to one, indicating similar mode shapes. Therefore it is correct to use mean values of f_n and ζ for the evaluation of the natural frequencies and modal damping in all the boundary conditions under test.
In general it was observed that the first mode shape of the brackets in clamped condition and clamped condition with accessories is clearly a bending mode, similar to the first bending mode of a fixed-free beam.

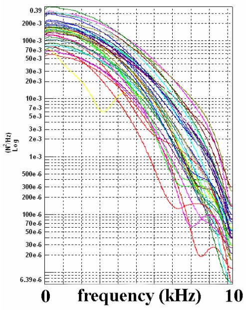
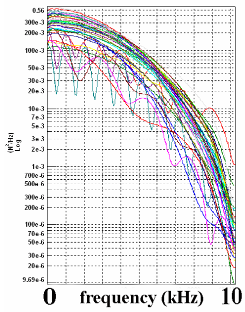
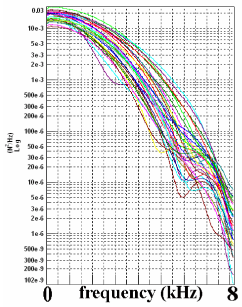
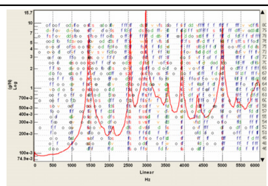
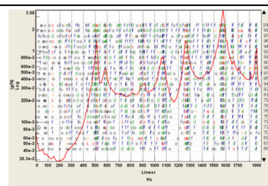
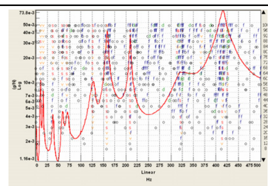
Bracket A	Free-free		Clamped		Clamped with accessory	
PSD OF THE EXCITATION FORCE						
BAND OF ANALYSIS	0-6000 Hz		0-1900 Hz		0-500 Hz	
STABILIZATION DIAGRAM by using LSCE solver and FRF SUM (in the band of analysis)						
Frequency and damping (mean value of LSCE and PolyMAX solver)	f_n [Hz]	ζ [%]	f_n [Hz]	ζ [%]	f_n [Hz]	ζ [%]
	1452	0,50	510	1,45	157	1,11
	2551	0,09	1258	0,67	209	0,82
	2965	0,13	1577	0,40	414	2,31
	3921	0,08			895	1,46
	5003	0,18			979	0,59

Table 5: results about bracket A

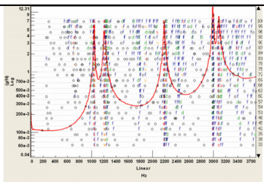
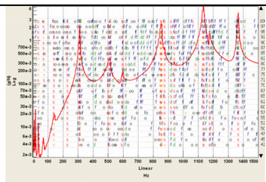
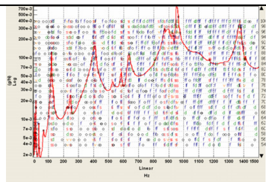
Bracket B	Free-free		Clamped		Clamped with accessory	
BAND OF ANALYSIS	0-3700 Hz		0-1500 Hz		0-1500	
STABILIZATION DIAGRAM by using LSCE solver and FRF SUM (in the band of analysis)						
Frequency and damping (mean value of LSCE and PolyMAX solver)	**f_n [Hz]**	**ζ [%]**	**f_n [Hz]**	**ζ [%]**	**f_n [Hz]**	**ζ [%]**
	1043	0,32	312	0,59	119	1,54
	1218	0,51	511	1,00	409	2,05
	2194	0,37	853	0,60	634	1,48
	2994	0,17	1134	0,58	898	3,16
	3088	0,39				

Table 6: results about bracket B

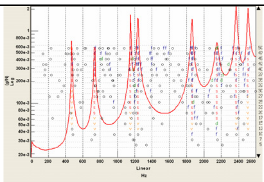
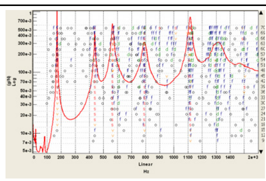
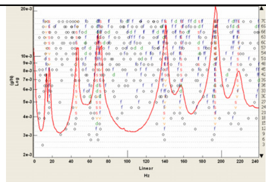
Bracket C	Free-free		Clamped		Clamped with accessories	
BAND OF ANALYSIS	0-2700 Hz		0-1600 Hz		0-240 Hz	
STABILIZATION DIAGRAM by using LSCE solver and FRF SUM (in the band of analysis)						
Frequency and damping (mean value of LSCE and PolyMAX solver)	**f_n [Hz]**	**ζ [%]**	**f_n [Hz]**	**ζ [%]**	**f_n [Hz]**	**ζ [%]**
	472	0,06	171	0,35	17	1,62
	735	0,10	439	1,00	46	2,77
	1151	0,09	565	1,02	68	1,94
	1235	0,13	791	1,07	72	1,92
			1116	0,09	141	1,18
					158	1,80
					194	1,09

Table 7: results about bracket C

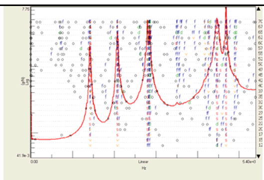
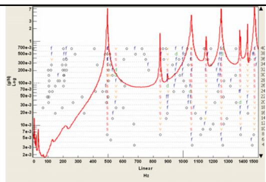
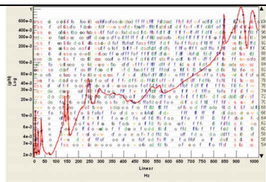
Bracket D	Free-free		Clamped		Clamped with accessory	
BAND OF ANALYSIS	0-5400 Hz		0-1500 Hz		0-1000 Hz	
STABILIZATION DIAGRAM by using LSCE solver and FRF SUM (in the band of analysis)						
Frequency and damping (mean value of LSCE and PolyMAX solver)	**f_n [Hz]**	**ζ [%]**	**f_n [Hz]**	**ζ [%]**	**f_n [Hz]**	**ζ [%]**
	1422	0,60	495	0,34	19,5	3,41
	2077	0,53	847	0,22	138	0,61
	2798	0,45	1055	0,30	152	0,61
	2857	0,25	1152	0,37	243	1,77
	4472	0,69	1252	0,17	290	2,55
	4686	0,20	1380	0,27	562	1,47
			1475	0,26	845	0,16
					923	0,26
					972	1,14

Table 8: results about bracket D

Figure 1 depicts the mean modal damping as a function of frequency, obtained by averaging the modal damping estimated by LSCE and PolyMAX solvers, for the four brackets in all the tested boundary conditions. The diagram shows three different zones that correspond to the different scenarios under study (free-free, clamped and clamped with accessories). The results obtained in free-free conditions show values of ζ between 0,06 and 0,71 % in the frequency ranges from 470 to 5000 Hz; the results in clamped conditions show values of ζ between 0,1 and 1,45 % in the frequency ranges from 170 to 1600 Hz; finally, the results carried out in clamped conditions with accessories show values of ζ between 0,2 and 3,4 % in the frequency range from 17 to 1000 Hz. It can also be noted that brackets with the same boundary conditions show similar values of modal damping, even if the brackets themselves have different shapes. In particular in the free-free scenario, the mean value concerning all the brackets is ζ = 0,30%, while in clamped boundary conditions the mean value is ζ = 0,63%. For the brackets with accessories the range is more extensive with mean value of ζ = 1,6%. Many problems arose during these last tests due to the complexity and non-linearity of the structures. In general, the more complex the structure is, the more uncertain the modal damping estimation is. Furthermore the accessories have different dimensions and mass, could produce different damping.
Figure 2 shows the values of the modal damping as a function of frequency obtained by interpolating the mean value of modal damping obtained for the four brackets in the same boundary condition. Such mean values can be used in dynamic analysis of new prototypes for which the hard prototype is not available yet.

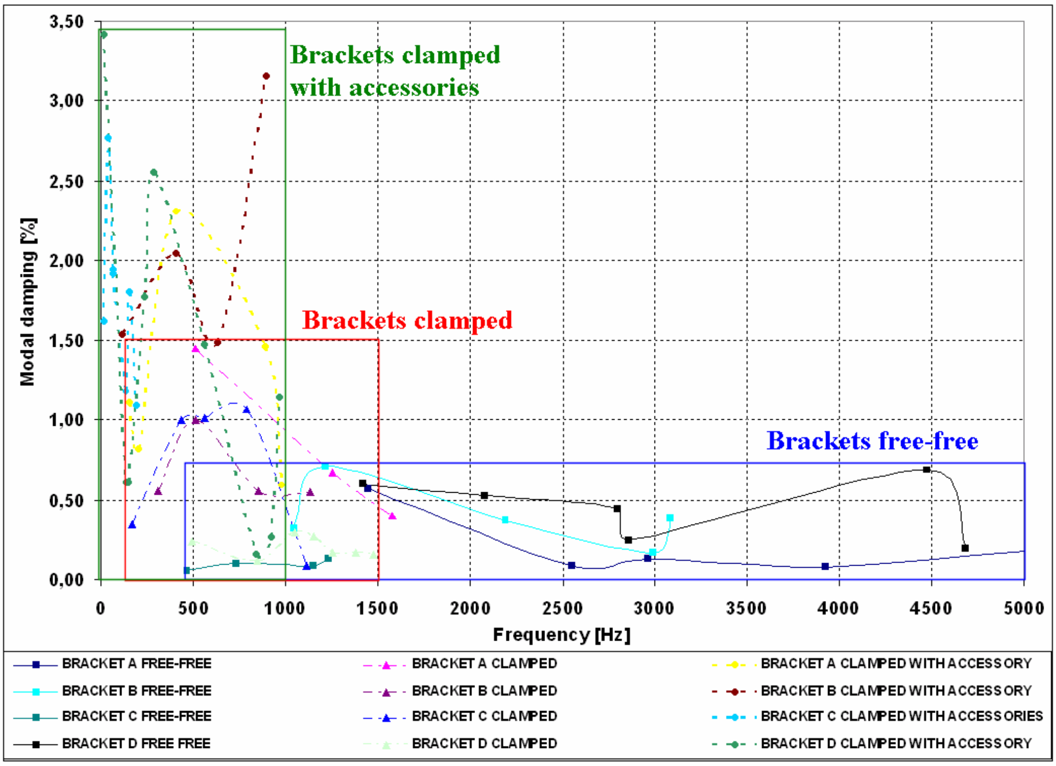


Figure 1 : modal damping [%] as a function of frequency for the four brackets in all the tested boundary conditions.

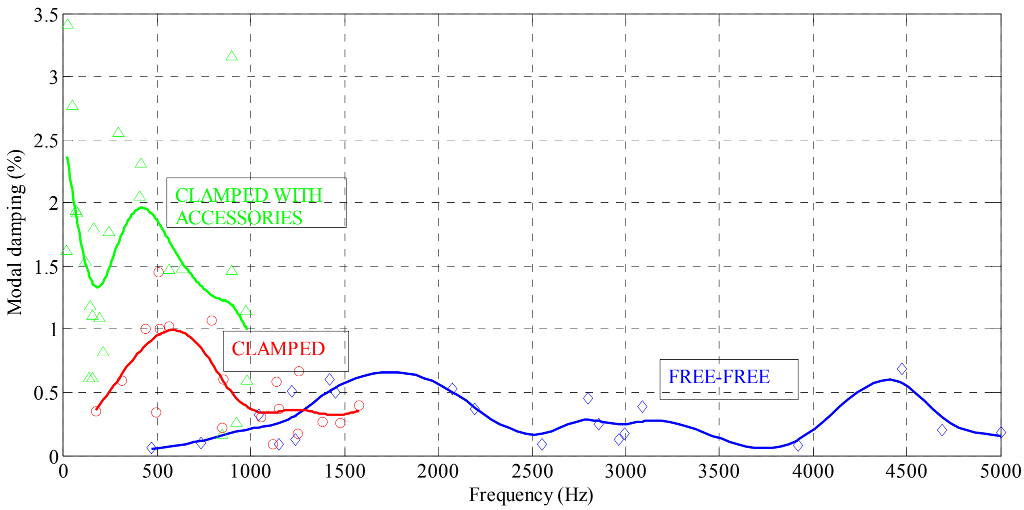


Figure 2 : interpolation of the modal damping values obtained for all the brackets in the three tested boundary conditions.

4. CONCLUSIONS

An experimental modal analysis program has been carried out on different typologies of brackets of diesel engines with different boundary conditions with the aim of collecting the damping values in a database, useful in the virtual prototyping phase for analysing the dynamic behaviour of new similar brackets for which the actual prototype is not yet available. In particular, experimental modal analyses have been carried out on four brackets with free-free, clamped and clamped-with-accessories boundary conditions. Then a database containing modal damping values for these typologies of brackets has been collected. The results show that the mean value of modal damping is 0,30 % in the free-free scenario, it is 0,63 % in clamped scenario and finally it is 1,60% in clamped scenario with accessories; however, in this last case the evaluation of the damping is more complex than in the others. Finally, the curves obtained by interpolating the modal damping values for the brackets under test have been shown. Such curves can be very useful in dynamic analysis of new prototypes of brackets for which no test data are available.

ACKNOWLEDGEMENTS

The authors wish to thank VM Motori S.p.A (Cento, Ferrara, Italy) and the engineers of this Company for co-operation and assistance in the collection of experimetal data. Part of this project has been carried out within the “Laboratorio di Acustica e Vibrazioni” (LAV), which is supported by the Regione Emilia Romagna - Assessorato Attività Produttive, Sviluppo Economico, Piano telematico - Fondi Obiettivo 2 (I).

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