Special test bench to investigate NVH phenomena of the clutch system

Dipl.-Ing. **A. Reitz,** Priv. Doz. Dr.-Ing. **J.-W. Biermann**
ika, Institut für Kraftfahrwesen Aachen, RWTH-Aachen

Dr. Eur.-Ing. **P. Kelly**
Ford Werke AG, Köln

Abstract

Difficulties in the investigation of driveline related NVH phenomena (Noise, Vibration and Harshness) arise from the fact that relevant levels of airborne and structure borne noises are superposed on by a number of different sources. Therefore it is helpful to develope special test benches that enable the investigation of single phenomena. Such a phenomenon is the so called clutch whoop. It occurs especially on vehicles with diesel engine during the process of disengaging and engaging the clutch. The effect is an unwanted tactile foot vibration with an accompanied disagreeable noise that sounds like the spoken word whoop. The lack of root cause analysis about this NVH issue lead to a series of investigations at the Institut für Kraftfahrwesen Aachen. Investigations showed that the clutch system is excited by torsional and axial flywheel vibrations. Torsional flywheel vibrations are caused by the 2nd engine order. A number of vehicle measurements highlight that the root cause of the axial flywheel vibration is a crankshaft bending excited by the 4th cylinder firing. The presented NVH test bench enables to separately investigate the axial and torsional aspects involved in the excitation process and therefore to gain a deeper understanding of the complex system behavior. Finally, the rig concept provides the ability to obtain good NVH design targets for clutch systems.

1 Introduction

The customers subjective perception of noise and vibration plays an important role for the assessment of modern vehicles. Therefore the vehicle development process is increasingly determined by Noise, Vibration and Harshness (NVH) orientated vehicle design.

An array of NVH phenomena are influenced by the clutch system. Useful tools to investigate NVH phenomena are vehicle measurements, test bench research and CAE analysis. For rudimentary investigations vehicle tests are inevitably used. But to gain deeper NVH knowledge for target setting, vehicle tests are not suitable due to the multitude of different excitation sources that occur at the same time. Therefore the contribution of individual sources on single phenomena frequently cannot be measured directly and thus cannot be consider separatly. The investigation on a special NVH test bench driven by an electric engine with a low noise and vibration levels, offers the possibility to focus directly on the real effects.

With this background special subsystem testbenches were constructed at the Institut für Kraftfahrwesen Aachen (ika) to get more insight in the NVH behavior of vehicle drivelines and to gain tools for the target setting procedure in an early project stage. The test bench presented in this paper is developed to analyse NVH phenomena of the clutch system.

2 NVH phenomena relevant for the clutch system

Fig. 2-1 shows frequently occuring NVH phenomena that are influenced more or less by the clutch system. Mostly all described phenomena are effected by both, vibration and noise aspects. On a test bench it is possible to investigate these aspects in more detail.

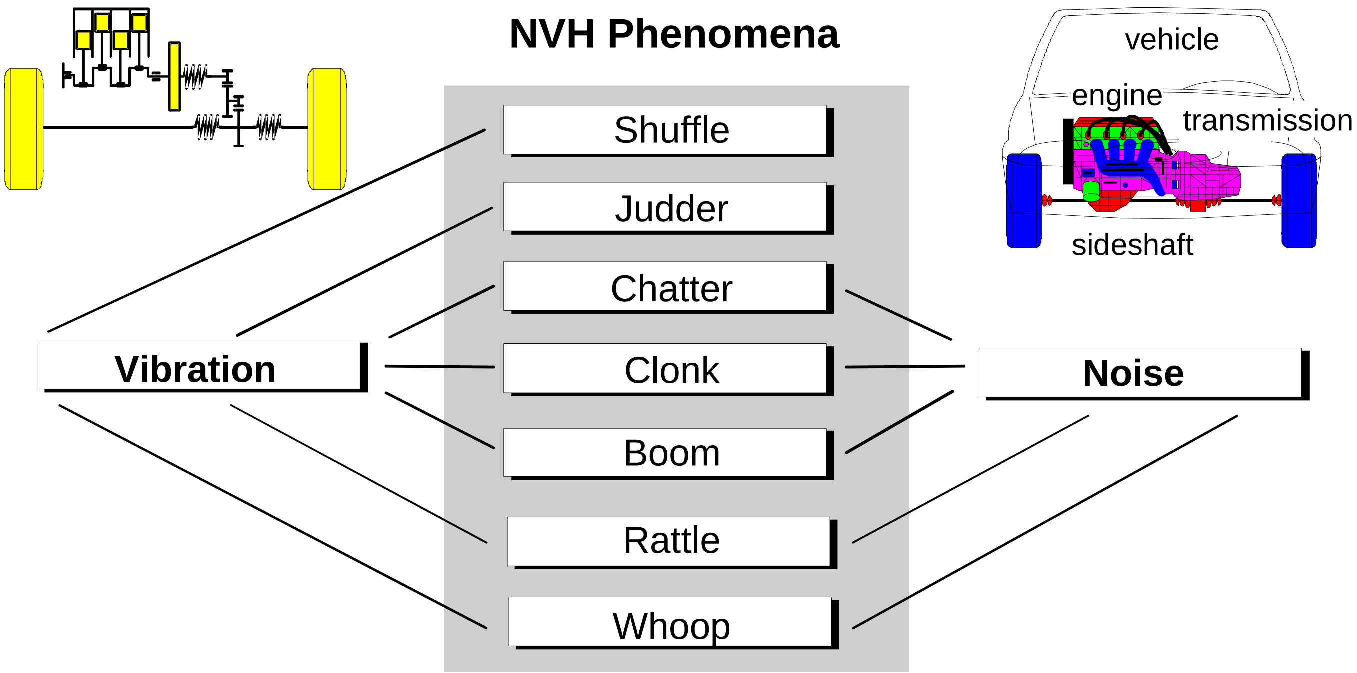

Fig. 2-1: Main NVH phenomena relevant for the clutch system

The shuffle is not a noise problem, it is a low frequent longitudinal vibration of the entire vehicle caused by load change with frequencies between 2 and 8 Hz. Shuffle is influenced by the clutch system in so far that the use of a twin mass flywheel has a significant effect. Investigations at the Institut für Kraftfahrwesen Aachen (ika) showed that a test bench simulation of the shuffle is not possible to date.

Clutch judder is also a low frequent fore and aft vibration of a vehicle in the frequency range 5 – 20 Hz, caused by the torsional vibrations of the driveline which occur during the clutch engagement process, usually in the drive away process especially on rear wheel drive vehicles with diesel engines. Take-off judder is essentially a total powertrain issue but can be influenced by frictional characteristics of the clutch [1, 2].

The Chatter phenomenon is decribed in a reference [3]. It is caused when a periodic torque change is generated in a slipping clutch whose natural frequency range is similar to that of the drivetrain dynamically separated from the clutch. The chatter is expressed as a vibration in the longitudinal direction of the vehicle and is transferred via the operating elements to the driver´s seat. It can also be perceived as interior noise. Chatter is devided in two different types. One is a self induced facing chatter caused by the friction coefficient. The second is a pressure induced chatter as a result of component deviations and axial vibrations.

Clonk is a hard metallic noise impulse of 20 to about 100 ms duration that can occur as a load change reaction. Other root causes are gear shifting or a quick clutch engagement. For the analysis of clonk on vehicles with frontwheel and rearwheel drive special test benches already exist at the Institut für Kraftfahrwesen Aachen. Investigations show that the characteristic of the clutch pre-damper stage or the use of a twin mass flywheel are important control factors for clonk [4].

Rattle and boom are also effected by the characteristic of a twin mass flywheel. Rattle is a broadband noise of the transmission caused by meshing of the loose gears induced by the 2nd or 3rd engine order. Boom is as well excited by engine orders and perceived in the vehicle interior when a body or cavity mode is covered. At the Institut für Kraftfahrwesen Aachen, investigations on idle rattle and in gear rattle are successfully carried out at specific transmission and driveline test benches [5].

The so called clutch whoop occurs during disengaging and engaging the clutch. The effects are low frequency pedal vibrations that increase during the pedal travel and are accompanied by an unpleasant noise [6]. The root cause of whoop is clarified by results from basic vehicle measurements described in the next section.

3 Vehicle investigations on the system behavior

Using five vehicles from different manufacturers the objective was to investigate the whole clutch system to increase the NVH behavioral understanding of whoop. Measurements were made at various engine speeds during disengagement and engagement with stationary vehicles and the transmission gear in the neutral position. The investigated engine speed range was between idle and 4000 rpm. The vibration of the clutch pedal and the release bearing were measured as well as the torsional vibration at the flywheel and the transmission input shaft.

Fig. 3-1 shows the clutch pedal vibration during disengagement and engagement at an engine speed of 2000 rpm. The left side presents the spectrum of the structure borne noise. At the beginning of the measurement the clutch is free. After 0.8 s the disengagement point (off point) is passed. Between 1.1 s and 1.2 s the clutch is fully disengaged before the clutch engages. The amplitudes of the vibration vary considerably with the position of the clutch pedal. The spectrum clearly shows the engine orders 1, 1.5, 2 and 2.5 increase during the disengagement process. The 2nd engine order is the dominant carrier frequency and reaches its maximum after the disengagement point.

At the right side of Fig. 3-1 the auto-correlation of the clutch pedal vibration is presented. The auto-correlation is a function that shows the correspondence of a signal to itself. The y-axis represents the time τ after that the signal is repeated. Thus the auto-correlation helps to detect the modulation frequency of a signal.

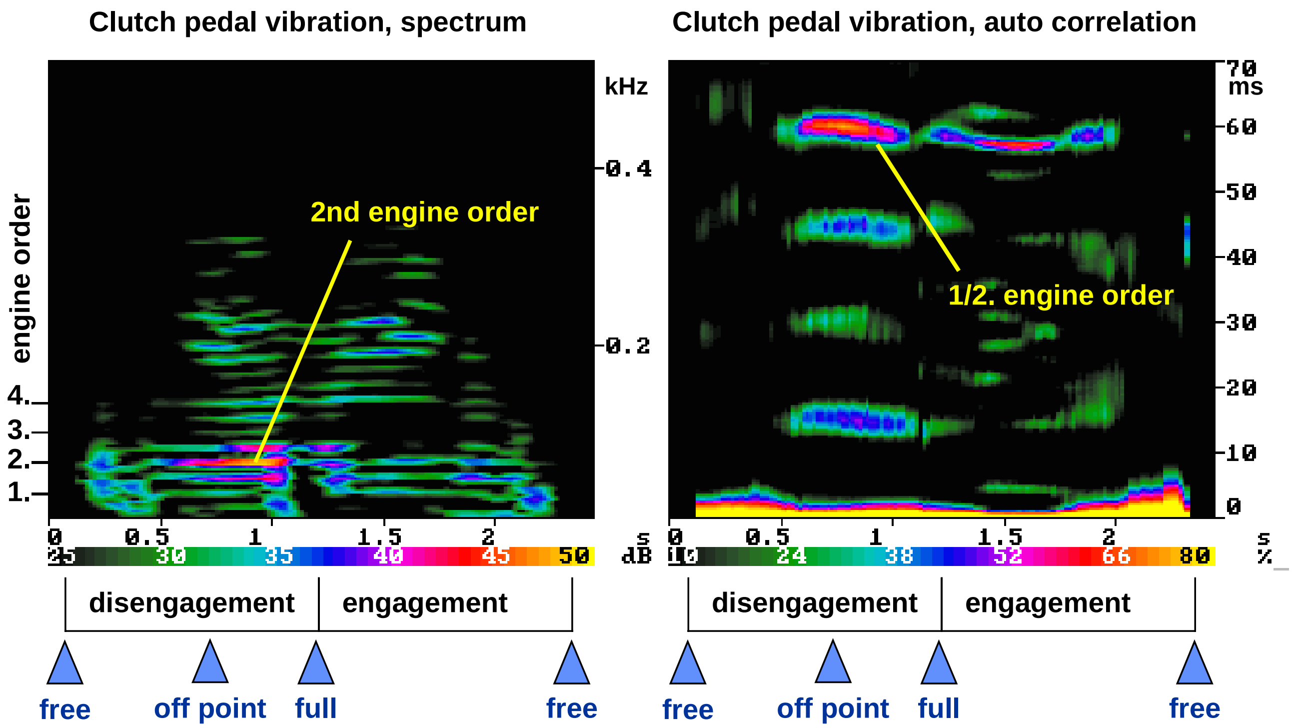

Fig. 3-1: Clutch pedal vibration during disengagement and engagement, n=2000 rpm

Here the signal is repeated after τ=60 ms. This is equivalent to a frequency of 16.7 Hz which is the half engine order at 2000 rpm. As a result the pedal vibration during the disengagement and engagement process is modulated with the half engine order.

The 2^{nd} engine order is excited by the torsional vibrations of the 4 cylinder diesel engine. In order to determine the root cause of the half engine order in the vehicle the flywheel modes are measured at three positions. Fig. 3-2 shows the result of this measurement at an engine speed of 2000 rpm.

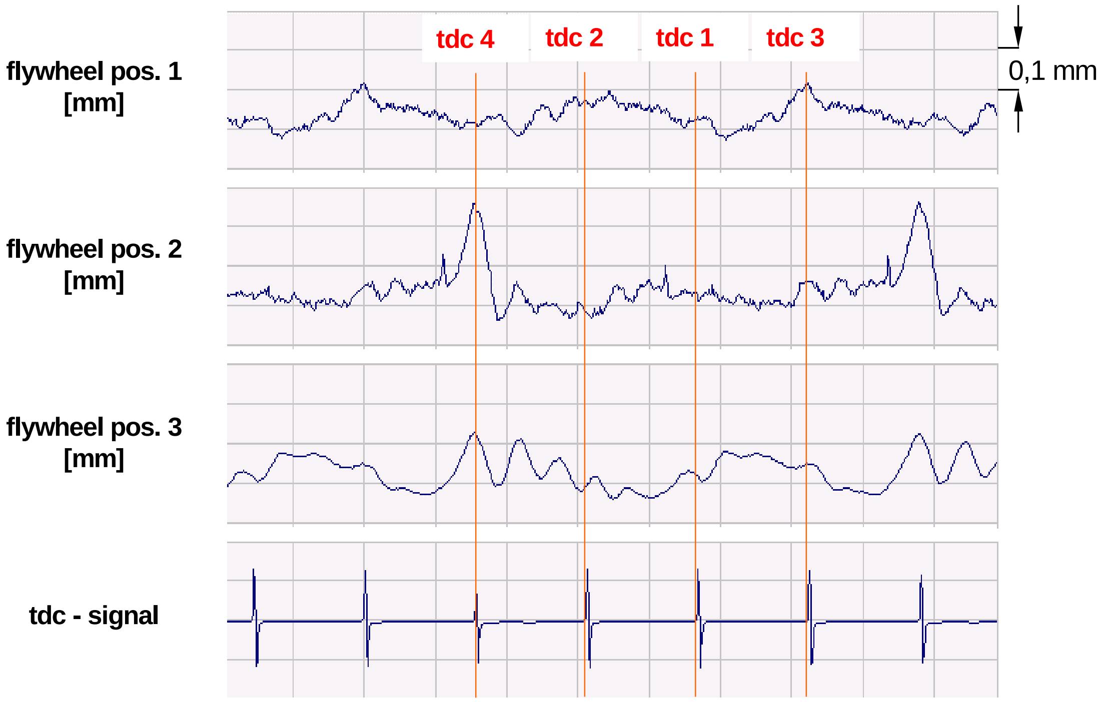

Fig. 3-2: Vehicle measurement of axial flywheel vibrations at n=2000 rpm, disengagement point

The figure is focused on a firing cycle of the 4 cylinder diesel engine at the disengagement point of the clutch.

It can be seen that the flywheel is excited mainly at the top dead centre of the fourth cylinder (tdc4). When the 4th cylinder fires a crankshaft bending is caused. The effect of this bending is an axial flywheel vibration that leads to an impulsive excitation to the clutch [7]. The maximal amplitudes of the measured vehicle amount to around 0.3 mm at flywheel position 2.

As a result of the vehicle measurements Fig. 3-3 shows the amplitude range of the critical flywheel modes.

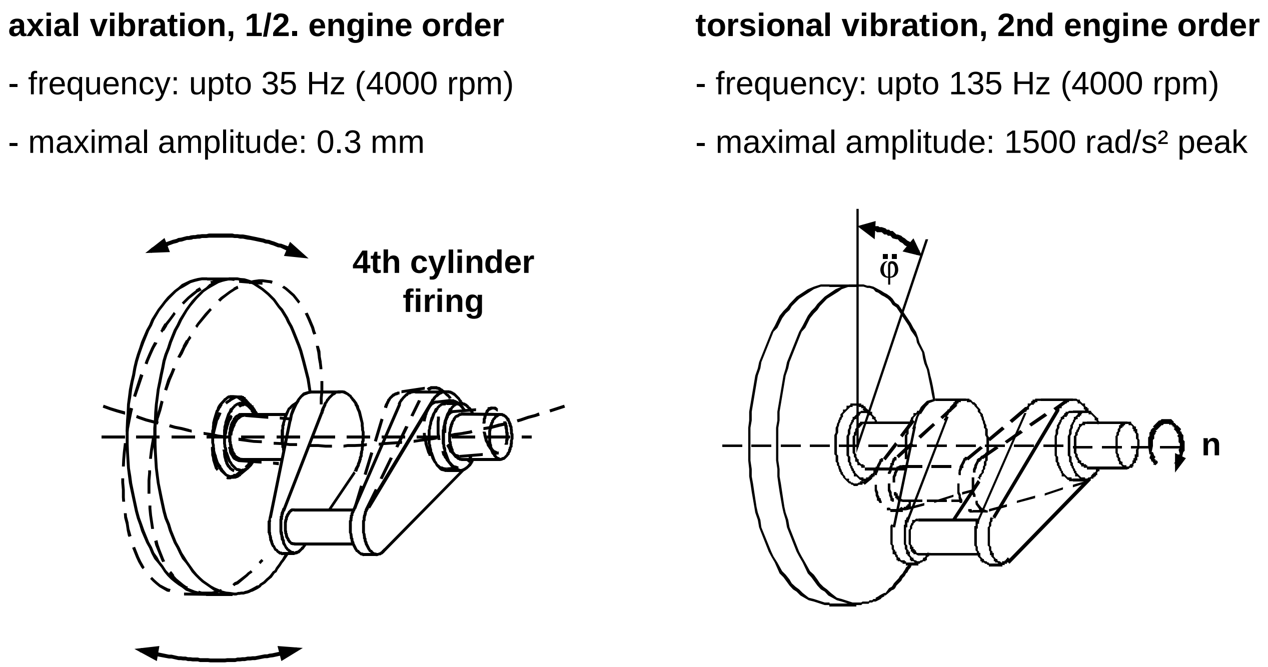

Fig. 3-3: Most critical flywheel modes, axial and torsional vibrations

The amplitude range of the axial flywheel vibration with half engine order is up to 0.3 mm. This depends on the bearing concept of the crankshaft. The 2nd engine order excitation with the transmission gear set in neutral position is influenced by the gas-forces and by the mass-forces which also depends on engine speed. At low engine speeds the gas forces are dominant. Around 2500 rpm the torsional vibrations have a minimum with amplitudes around 200 rad/s² peak. At higher engine speeds the 2nd engine order is predominantly influenced by the mass forces with torsional vibrations up to 1500 rad/s² peak (at 4000 rpm) at the measured vehicles with diesel engine.

The vehicle measurements demonstrate the complexity of flywheel excitation and its transfer to the clutch pedal. Investigations on a test bench will help to gain a deeper understanding also for CAE modeling.

4 Test bench to simulate clutch phenomena

The knowledge gained from the vehicle measurements is the basis to design the layout of the test bench. On the test bench the separate investigation of axial, rotational and torsional vibration aspects should be possible. Further demands are mentioned below:

- Simulation of clutch pedal vibration in idle position with constant engine speeds during engaging and disengaging the clutch
- Assembly of the whole clutch system to investigate the transfer behavior
- A low noise and vibration drive by using an electric motor

Fig. 4-1 shows the concept of the test bench. The clutch system is assembled completely with transmission, hydraulic system and pedal box. An electric motor is used as power input. The axial flywheel vibrations are excited by a connecting rod and an eccentric shaft. This shaft is powered by a belt drive with a speed ratio of 2:1.

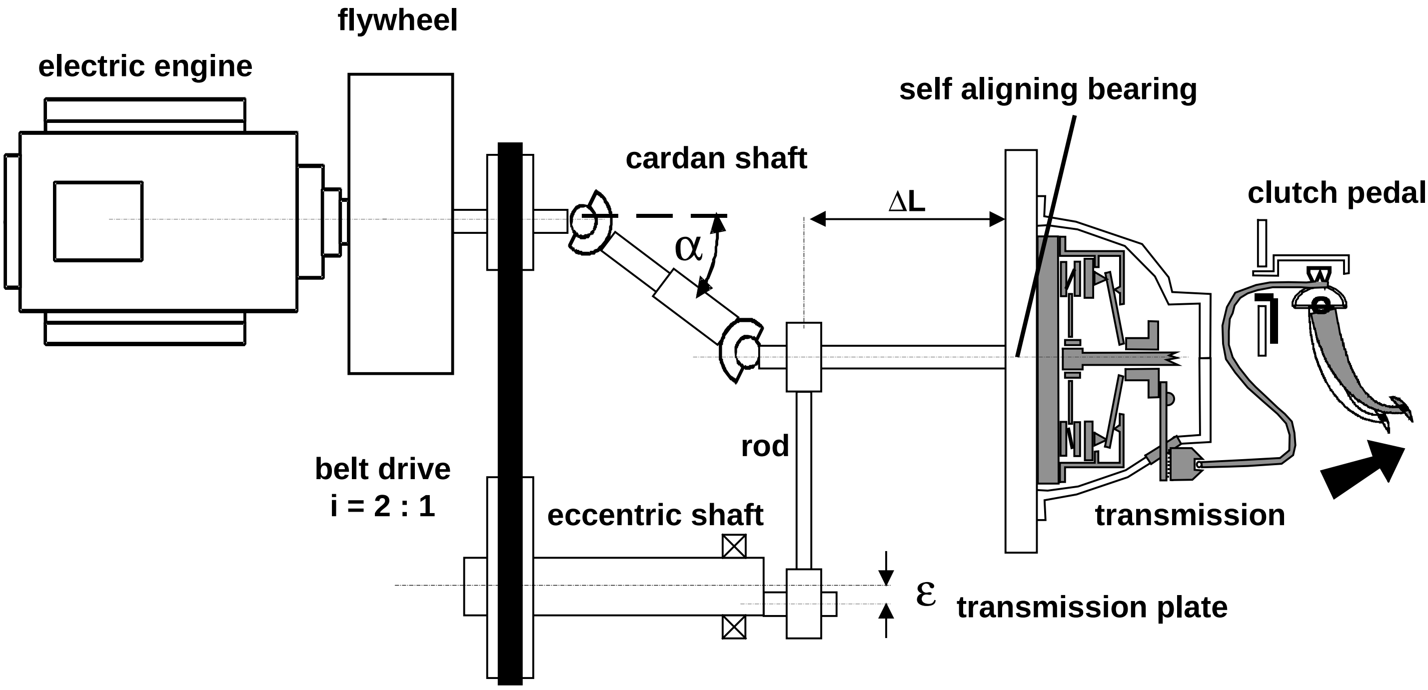

Fig. 4-1: Concept of the test bench to investigate the clutch system behavior [8]

A self-aligning bearing mounted in the transmission plate allows a rocking motion of the flywheel. The distance ΔL between self aligning bearing and connection rod influences the amplitude of the axial flywheel excitation. If the crankshaft deflection of one engine type is known, the axial excitation of the flywheel can be adjusted in the right way. The bigger the distance ΔL the smaller becomes the amplitude.

To simulate the 2nd engine order a cardan shaft is used. Rotating with a bending angle α cardan joints produce torsional vibrations with a 2nd order. Fig. 4-2 (a) illustrates this effect. If two cardan joints are assembled in such a way that both have the same angle and both joints rotate in one surface, the irregulations are neutralised and the output has no torsional vibration. This clarifies Fig. 4-2 (b). If two cardan joints are assembled 90° twisted then the 2^{nd} order is amplified (Fig. 4-2 (c)). This effect is used for the presented test bench. The relation between the angles φ_{11} and φ_{22} follows Eq. 4-1:

$$\varphi_{22} = \arctan\left(\frac{1}{\cos^2\alpha}\cdot\tan\varphi_{11}\right) \qquad \text{Eq. 4-1}$$

Apparently φ_{22} and so also the torsional vibration depend on the bending angle α. Thus by a variation of this angle the amplitude of the 2^{nd} order can be increased continuously at the test bench. The excitation frequency can be regulated by adjusting the speed of the electric motor.

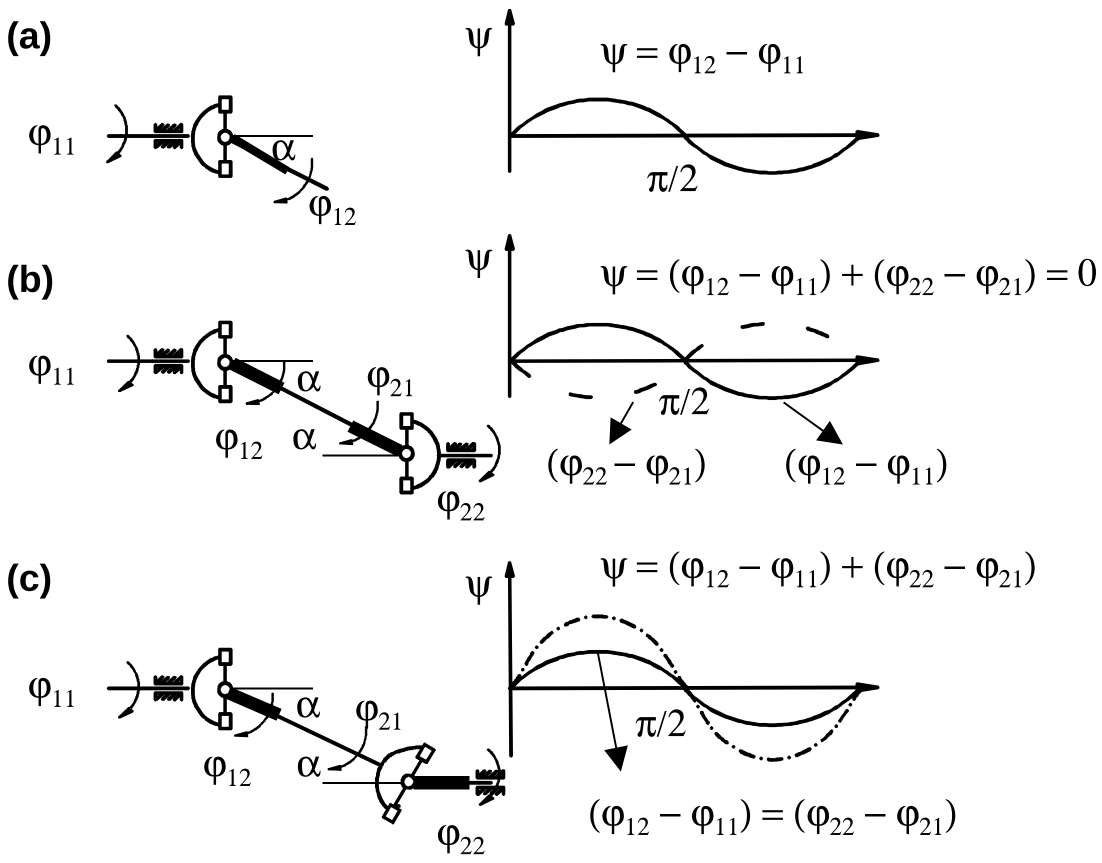

Fig. 4-2: Effect of torsional vibrations induced by cardan joints [9]

Fig. 4-3 shows the actual test bench. The clutch system is tested together with the gearbox to simulate realistic conditions. The pedal box is fixed on a console beside the bench foundation.

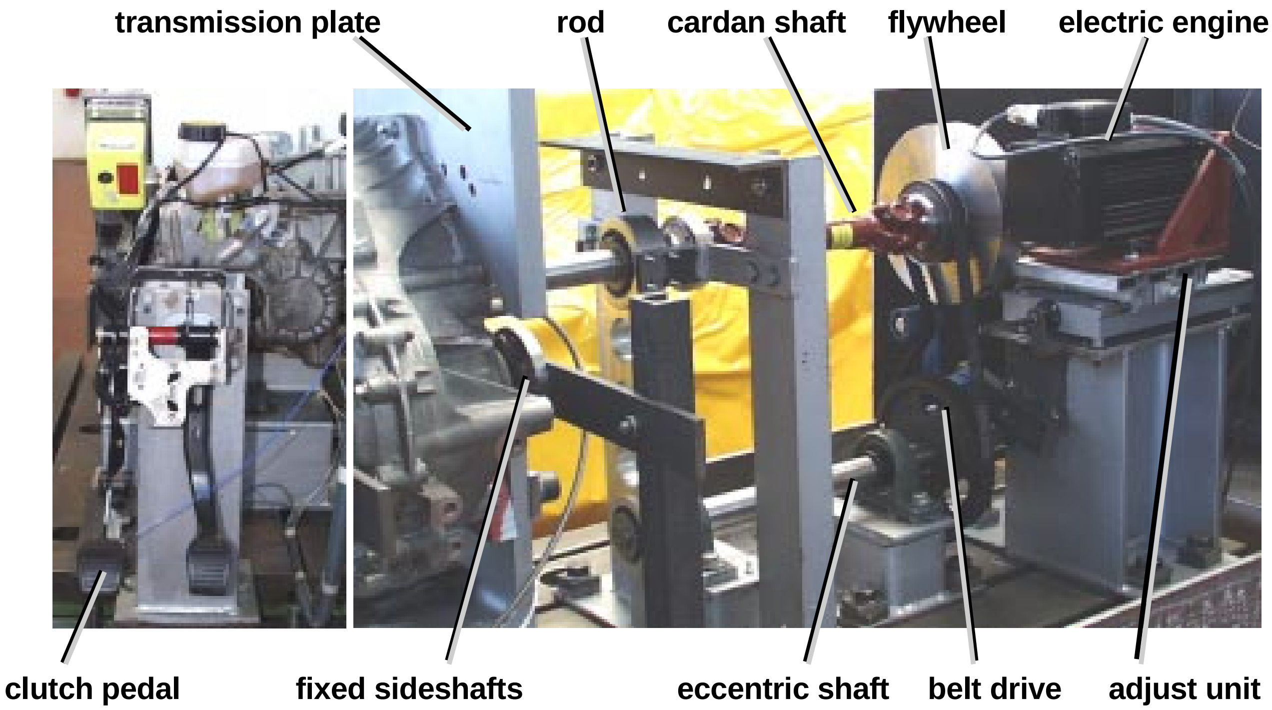

Fig. 4-3: Test bench for clutch system investigations

This layout enables to determine that part of pedal pad vibration, which is transfered by the hydraulic actuation system. For a better repeatability the clutch pedal is applied by a hydraulic cylinder.

It is not necessary to simulate different loads because the tests are done without gear so the sideshaft system is fixed like in a vehicle at rest. The oil temperature is regulated by a hot-air blower.

With the described test bench concept it is possible to simulate each kind of excitation individually for detailed analyses. The following described conditions can be investigated:

- rotation (800 to 4000 rpm)
- axial flywheel vibration (±0.0 to ±0.5mm)
- axial flywheel vibration + rotation , 1/2. engine order
- torsional vibration, 2nd engine order (0 to 1500 rad/s²)
- axial excitation (1/2. engine order) + torsional vibration (2nd engine order)

Fig. 4-4 shows the sensor positions to measure the reactions of the clutch system.

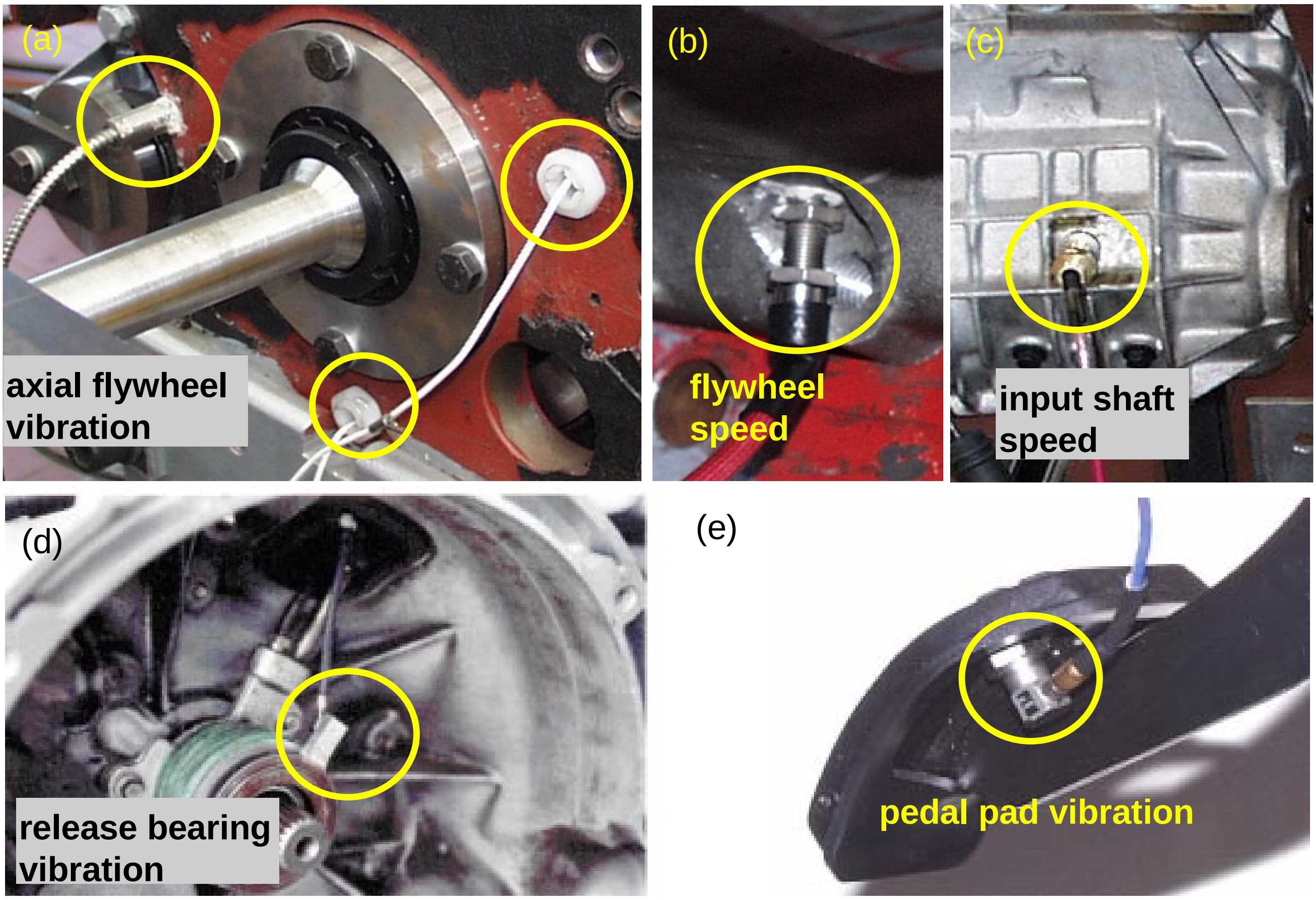

Fig. 4-4: Sensor positions for the test bench investigations

The axial movement of the flywheel is measured at three positions with eddy-current sensors. The torsional vibration of the flywheel and the transmission input shaft are measured by the rotation analysis system of the ROTEC GmbH München. The used system enables to measure six torsional vibration channels and 24 analogues channels simultaneously.

Structure-born noises at the release bearing and the clutch pedal pad give information about the reactions of the system.

5 Results of testbench investigations

The given examples clarify the possibility of analysing the process in the system by means of the testbench. Fig. 5-1 represents the effect of the torsional vibration on the clutch pedal vibration as a function of the pedal travel when disengaging. The flywheel speed is 2000 rpm. While disengaging the clutch pedal vibration increases. If the individual curves are compared, it becomes clear that the amplitude of the 2nd engine order has an influence on the pedal vibrations. The higher the torsional excitation of the flywheel, the bigger the acceleration of the pedal. At 1200 rad/s² the values are significantly higher than at 0 rad/s².

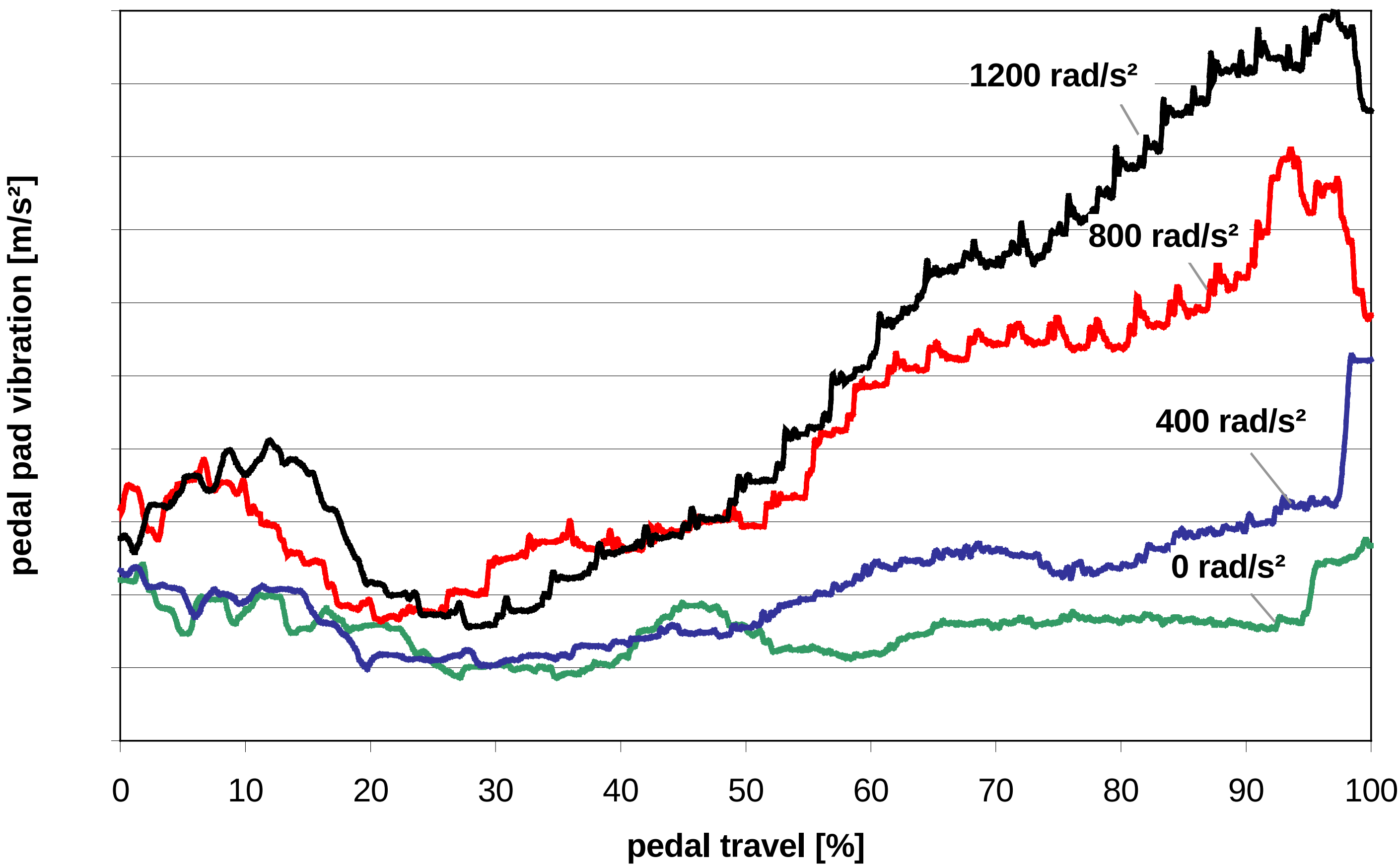

Fig. 5-1: Effect of the second engine order on the pedal pad vibration at 2000 rpm

This test is not possible in the vehicle because the excitation of the clutch pedal is influenced by other sources and transfer paths. Therefore the analysis on the test bench is also a benefit for CAE-simulation.

In the same way a comparison between different amplitudes of axial excitation is possible. The lowest clutch pedal vibration is possible without axial or torsional excitation when the flywheel rotates continuously. In this condition the measured values predominantly are produced through slip stick effects in the friction lining. The pedal vibrations are about twice as high when the flywheel rotation is superposed by an axial excitation. The level is equivalent to a torsional excitation with 400 rad/s² peak. In vehicles with diesel engine the 2nd engine order excitation at 2000 rpm is also arround 400 rad/s². The minimum vibrations when gas-forces change to mass forces are even lower. This illustrates that at conditions with a very low torsional excitation the axial flywheel vibration is dominant for the clutch NVH. The higher the amplitudes of the 2nd engine order for example at higher engine speeds the more the clutch system behavior is effected by torsional aspects. With this knowledge gained from the excitation experiment it is possible to propose targets for a reduction of the pedal vibration. The values of these targets can be gained from the test bench measurements.

In parameter studies a tuning of the clutch system components is possible. As an example Fig. 5-2 shows the result of a test bench and a vehicle measurement to evaluate a clutch design manufactured to reduce pedal vibrations. To protect the confidentially requirements, the different designs are referred to as letters A to C. Also the conduction of the parameter study is not explained.

In Fig. 5-2 it is easy to see which design leads to the best improvement. The highest pedal vibration is measured with design A which is the original clutch system. With design B and C clearly lower accelerations are obtained. By tuning the eigen frequencies regarding each excitation kind the favorable gained design is C.

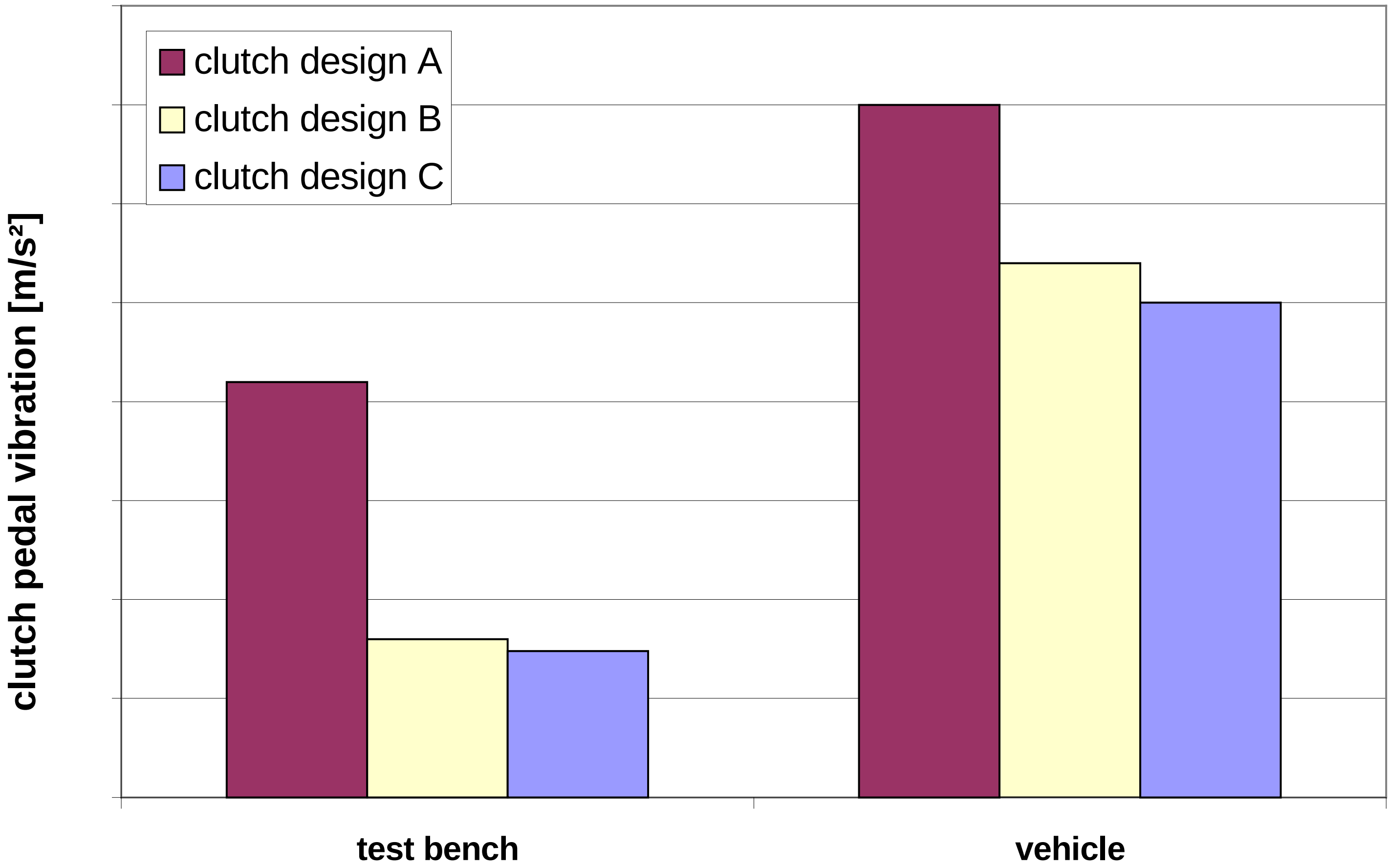

Fig. 5-2: Improvement effect of different clutch designs, testbench and vehicle measurement

In the vehicle the test bench tendencies are repeated. Only the amplitudes of the clutch pedal vibration in the vehicle are higher than on the test bench, because in the vehicle the pedal vibration is also covered by additional sources and transfer paths. On the test bench the transfer from the flywheel along the hydraulic system to the clutch pedal is measured without masking effects.

It can be seen that the test bench presented in this paper enables to determine the clutch systems NVH behavior during disengagement and engagement in more detail. The behavior of the clutch system in a real vehicle can be predicted in general on a test bench by measuring the characteristic for each kind of excitation. So it is possible to find a way to reduce the vibration and noise and to propose good NVH targets.

References:

[1] Drexl, H. J.
"Clutch judder – causes and counter measures"
Technische Konferenz SITV 1990

[2] Rabieh, E. M. A., Crolla, D. A.,
"Coupling of driveline and body vibrations in trucks"
SAE Conf. Trans. on Int. Trucks and Bus Meeting and Exposition, Vol. 1203
Detroit, 1996

[3] Albers, A., Herbst, D.
"Chatter – Causes and Solutions"
6th LUK Kolloquium 1998

[4] Hagerodt, B.
”Untersuchung zu Lastwechselreaktionen frontgetriebener Personenkraftwagen”
Dissertation am Institut für Kraftfahrwesen Aachen
Schriftenreihe Automobiltechnik, Forschungsgesellschaft Kraftfahrwesen (fka)
Aachen, 1998

[5] Reitz, A., Biermann, J. W., Kelly, P.
”Special testbenches to investigate driveline related NVH phenomena”
8th Aachen Colloquium, ”Automobile and Engine Technology”, 1999

[6] Kelly, P. and Rahnejat, H.
"Clutch pedal dynamic noise and vibration investigation"
International Symposium on Multi-body Dynamics – Modelling and Simulation Techniques
MEP, Bradford, UK, Mar 1997

[7] Kelly, P., Biermann, J. W., Hagerodt
”Clutch pedal dynamic noise and vibration investigation using Taguchi methods”
6th Aachen Colloquium, ”Automobile and Engine Technology”, 1997

[8] Reitz, A.
”Investigation of Clutch Whoop on a test bench”
Report 7482 (not published)
Institut für Kraftfahrwesen Aachen (ika)
Aachen, 1999

[9] Duditza, F.
”Kardangelenkgetriebe und ihre Anwendungen”
VDI-Verlag GmbH
Düsseldorf, 1973